

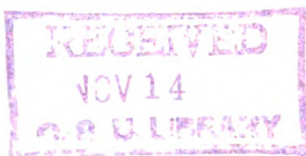
DEPARTMENT OF THE INTERIOR

UNITED STATES BUREAU OF MINES
== JOHN W. FINCH, DIRECTOR ==

INFORMATION CIRCULAR

SAND AND GRAVEL EXCAVATION;

PART 4: CAR AND LOCOMOTIVE HAULAGE;
HOIST AND ROPE HAULAGE;
REMOTE-CONTROL HAULAGE;



BY

J. R. THOENEN

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ERRATA SHEET

I.C. 6856, Issued Oct., 1935

Page 149 now reads

$$T = 4 \times \frac{\pi d^2 PS}{4\pi D} = \frac{\pi d^2 PS}{D}$$

should read

$$T = 4 \times \frac{\pi d^2 PS}{4\pi D} = \frac{d^2 PS}{D}$$

Page 153, seventh word, second line should be mean instead of men.

Page 157, formula at bottom reads

$$\frac{5730}{15} = 19.54 \text{ miles per hour}$$

should read

$$\sqrt{\frac{5730}{15}} = 19.54 \text{ miles per hour.}$$

Page 175, formula reads

$$S = \frac{\sqrt{8DT}}{W} = \frac{\sqrt{8 \times 1 \times 10000}}{488} = 164 \text{ feet}$$

should read

$$S = \sqrt{\frac{8DT}{W}} = \sqrt{\frac{8 \times 1 \times 10000}{488}} = 164 \text{ feet.}$$

INFORMATION CIRCULAR

DEPARTMENT OF THE INTERIOR - BUREAU OF MINES

SAND AND GRAVEL EXCAVATION - PART IV:¹ CAR AND LOCOMOTIVE HAULAGE; HOIST AND ROPE
HAULAGE; REMOTE-CONTROL HAULAGE

By J. R. Thoenen²

INTRODUCTION

This circular is part IV of the third paper (entitled "Sand and Gravel Excavation") of a series summarizing the technical problems involved in the production and preparation of sand and gravel. Part I discussed the use of power shovels, draglines, and excavator cranes; part II discussed the use of power scrapers, slackline cableway excavators and hydraulic monitors; and part III discussed the use of hydraulic, clamshell, ladder, and dipper dredges. This circular discusses haulage equipment in general, particularly the use of industrial cars with locomotives or hoists as the motive power. Part V, to follow, will discuss the use of other types of haulage equipment.

HAULAGE EQUIPMENT

Haulage equipment as used at sand and gravel pits may be divided into three classes: (1) That used to receive overburden and transfer it to the waste dump; (2) that used to receive sand and gravel from the digging unit and transfer it to the treatment plant. Where the digging unit is also the treatment plant the haulage unit transfers from the digger to points of distribution; (3) that used to receive sand and gravel from the plant and transfer it to storage, to recover it from storage and return it to the plant for further treatment, or to recover it from storage and transfer it to shipping equipment.

Haulage equipment in the first class has been discussed in a previous paper but not in as great detail as herein. A study of the function of haulage equipment in the third class belongs in a paper designed to follow this and hence will not be discussed here. This paper will deal directly with the function of haulage equipment in the first and second classes, namely, that used to handle overburden, receive sand and gravel from the digger, and transfer it to plant or points of distribution.

Except for the power scraper, cableway excavator, hydraulic dredge, and hydraulic giant, excavator units themselves do no hauling, and at times even these units depend on supplementary haulage equipment. With the exceptions mentioned, excavation units depend upon the haulage unit for efficient operation.

Haulage units cannot transfer material that has not been dug. Therefore haulage equipment is subordinate but complementary to excavation equipment. This requires that the haulage unit must be carefully coordinated to serve the digger properly. The first and most important consideration is coordination in capacity. Capacity does not mean the size of the vehicle but the ability of the unit to receive the required tonnage from the digger and to deliver it over the necessary haulage route to the plant or distribution point in the re-

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quired time. Regardless of the type of haulage unit selected, the problem of capacity is of prime importance, and on its correct solution depends the proper balance of operation in delivering material to the treatment plant.

Selection of the proper type of haulage unit depends upon other factors, such as the type of excavator, the mining method, the surface topography, the character of the material mined, the power available, and the past experience and preference of the operator.

The types of haulage units used with various types of excavators are as follows, preference being shown by order of mention.

1. With power shovels and draglines: Industrial cars and locomotives; autotrucks; industrial cars and hoists; conveyor belts.

2. With cranes: Industrial cars and locomotives; autotrucks.

3. With power scrapers and cableway excavators: These types usually do their own hauling, but each may supplement the other or a like unit. In rare instances each may use cars or autotrucks.

4. With hydraulic dredges: Pumps and pipe lines; barges; power scrapers; cableway excavators.

5. Clamshell, ladder, and dipper dredges: Barges.

6. Hydraulic giants: Sluices; pumps and pipe lines.

The effect of the method of mining on the haulage equipment to be used will closely parallel its effect on the excavating unit, since the method used is generally planned to suit the unit. However, varying methods may change the order of preference as follows:

1. High banks may require conveyor belts in preference to either cars or autotrucks.

2. Thin deposits of wide area may require autotrucks in preference to other types.

3. Deep, dry pits may require hoists in preference to locomotives.

4. Thin beds in wet pits may require barges in preference to pipe lines.

5. River deposits may require barges in preference to pipe lines.

6. Lake and ocean deposits may require self-propelled dredges in preference to a separate transportation unit.

The character of the surface may be such as to prohibit the expense of establishing roadbeds for cars operated on tracks whereas autotrucks could be used. For very rough country a combination of cars or trucks with an aerial tram may be necessary. Hoists may be necessary to raise or lower the material to the plant.

Wet pits may require power scrapers or cableway excavators in conjunction with cars or trucks or even aerial trams. The supplementary units are seldom required unless the pit is considerable distance from the plant.

High river banks may necessitate barge haul even though the hydraulic dredge is located close to the plant. Swift river currents may prevent the use of floating pipe lines and necessitate barge haul. To avoid the effect of swift currents pipe lines are sometimes submerged and laid on the river bottom.

The character of the material may have an important bearing on the type of haulage unit selected. An excess of large boulders requires heavy construction of the haulage unit and may require larger units.

An excess of clay may require pumps and pipe lines in preference to barge haul in order to wash the material more thoroughly before it enters the plant. If many large boulders occur with the clay the pipe line may have to give way to barges because of excessive wear. Wet material from pit or marine deposits may require structural differences in the haulage unit which would not be necessary with dry material. Well-rounded sands and gravels may be pumped easier than sharp-edged or flat particles. The shape of individual particles may differentiate between pipe lines or barge haul.

The type of power available has a direct bearing on the type of haulage unit. With cheap current electric locomotives of the trolley or storage-battery type may offer the best

possibilities. Under the same conditions cheap power may warrant transportation with one or more booster pumps rather than barge haul. Scarcity or high cost of coal may require that gasoline or Diesel locomotives or hoists be used instead of steam units. High power costs may indicate that conveyor belts are best in spite of otherwise adverse conditions. Lastly, the past experience and preference of the operator will largely influence the type of haulage unit selected.

It is evident that the factors affecting the type of haulage unit to be used are largely local and vary with individual deposits. Therefore, the choice of type is individualistic and impossible of standardization.

So many designs are available in the different types of haulage equipment that construction details can be discussed only generally. The advantages or disadvantages of general variations in type will be presented, but the principal discussion under the several types will relate to their capacity and methods of calculating capacity.

Since the greatest efficiency is obtained when excavating equipment is operated continuously at or near working capacity, the haulage unit must assure equal or greater capacity under adverse haulage conditions. The determination of capacity in the haulage unit involves such factors as the length of haul; vehicle size and number; motive power and speed; and conditions as to track gage, curves, and grades.

Vehicle size and number, motive power and speed, and track gage and curves require no explanatory definition. Length of haul is defined as the distance between excavator and dump measured over the haulage route in the direction of the loaded haul. In circular or elliptical haulage routes the loaded haul may temporarily exceed the empty return, but this is usually equalized by reverse conditions as the excavator moves. Length of haul in such cases is usually considered as the average loaded haul between excavator and plant. With conveyor belts, aerial trams, and barges the loaded haul and empty return are equal, but with pipe lines there is, of course, no return.

Grades are considered favorable or adverse, depending upon whether the loads move downhill or uphill.

Industrial Cars

Industrial cars operating on steel rails are probably the most popular form of pit haulage used in connection with mechanical excavators. Built in a multiplicity of designs and sizes they can be adapted to any production requirements within the physical limitations of this type of transportation. They have the common disadvantage of requiring rails and ties and a maintained roadbed. This limits their flexibility to a certain extent, although the provision of rails or track in short portable sections reduces the time lost in shifting. As a general rule the heavier the car the greater care must be exercised in roadbed construction and maintenance.

Cars may be pushed by hand or hauled by animals, locomotives, or hoists, or they may have their own motor equipment. When pushed by hand they are handled as single units and seldom exceed a payload capacity of 3 tons. Handled by horses or mules they may be coupled in trains of two or more cars, but the gross weight of trains seldom exceeds 10 tons. For locomotive or hoist power, cars may range in payload from a few tons to standard railway equipment of 100 tons each. Cars may be handled singly or in trains by locomotives, but hoists more often are limited to single cars. The number of cars in a train for locomotive haul is limited by the size and power of the locomotive and the grade and curvature of the track. Self-propelled cars are operated as single units but may range up to 50 tons or more in payload.

While track gage usually increases with the gross weight of the car or train, the increase is not always proportionate. Cars of relatively large capacity have been designed and

used successfully on narrow-gage tracks, but such requirements are usually caused by some specific local condition.

The ability to use any number of cars in a train within the limit of the size and power of the locomotive permits a wide flexibility in the capacity of the system in that increased tonnage requirements may be accommodated by adding one or more cars to a train. A further increase may be met by replacing the locomotive with a larger unit or by the addition of another locomotive and train.

The substitution of hoists for locomotive power is usually caused by excessive grades. Hoists are used in two ways - as a power unit to pull loaded cars up an incline and as a simple brake mechanism to lower loads on an incline. In either case the hoist may be of the double-drum type with two cables or the single-drum type with one cable, one end of which is payed off the top of the drum while the other end winds on at the bottom. When hoisting loads on one cable or one end of the single cable the other is attached to and lowers the empty car. When used as a brake mechanism the descending loads furnish power to bring up the empty.

Where inclines are necessary, some form of pit haulage is usually employed to bring the loaded cars from the excavator to the foot or top of the incline. This initial haul may be either by hand, animal, locomotive, or autotruck. If the latter is used the truck usually dumps its load to a hopper from which the car or skip on the incline is loaded.

As a comprehensive discussion of the structural details of pit cars would fill a volume it cannot be included in a paper such as this. The reader is referred to manufacturer's catalogs for such details.

Other things being equal, the length of haul has a direct bearing upon the capacity of a haulage unit using cars and locomotives or cars and hoists. For the same size and number of cars the capacity will diminish as the length of haul increases, although not in direct proportion. In computing the effect of haulage length one must consider the completed round trip as a unit, and this must be split into several component parts such as the time required for loading, hauling to plant, dumping, returning empties to digger, and waiting at digger or dump. The loading, dumping, and waiting time for similar units in a single pit are apt to be fairly equal per haulage cycle, irrespective of whether the length of haul is 100 or 1,000 feet. Moreover, the time required to start and accelerate as well as to brake and stop a trip are apt to be fairly uniform and independent of the length of haul. The time required to haul loads is apt to be greater than that to return empties, even though both in- and out-haul distances are equal.

Long hauls ordinarily exceed shorter ones in the initial construction cost per foot because they are usually built to remain in place for longer periods and hence are better alined and ballasted. Operating costs per ton-mile on long hauls are usually less because speeds are higher and maintenance expense is less. Short hauls on frequently shifted tracks, which cannot be heavily ballasted or are crooked and poorly alined as to curves, require heavy maintenance expense.

Other things being equal, on a long haul it is usually more economical to use fewer large cars and faster locomotives rather than the same load capacity in small cars.

Large cars are as a rule easier to load by mechanical equipment than small cars because less time is required to spot the bucket or dipper over them so as to prevent spillage. While more time will be required to load a large car, the loading time per ton of payload may be reduced. However, spotting is as much a function of the locomotive as the digger.

Low cars are easier to load than high cars, since the bucket or dipper need not be raised so high to clear the sides. Low cars also lower the center of gravity and thereby reduce the derail hazard.

Generally long cars with long wheel bases are more difficult to derail and likewise are harder to replace once they jump the track. They also require curves of greater radius. However, this depends upon whether the car has 4 or 8 wheels and whether the trucks are rigidly connected to the car body.

Large-capacity cars with the same number of axles offer less frictional resistance per ton of payload than small cars. The payload in a large car is concentrated and supported on 4 to 8 journals or bearings, while the same load hauled in smaller cars would be distributed in more cars having from 4 to 8 bearings for each car, each of which contributes frictional resistance. However, the type of bearing may reverse this condition. For instance, a 50-ton car on plain cylindrical bearings in poor condition will afford more resistance than two 25-ton cars fitted with roller bearings.

Larger cars reduce the number of cars in a train for the same payload and hence reduce dumping time per trip, also loading time if the time required to move the train to spot the empty, is more than the excavator time cycle. Shorter trains are also easier to manipulate over the ordinary poorly ballasted pit tracks.

Side-dump cars are commonly built with a center hinge so that they may be dumped to either side. These provide more operating flexibility but usually afford less running stability than those built to dump to one side only. Where the haulage route is a circle or ellipse and the equipment moves continuously in one direction, cars which dump to one side only have often been found to be time savers. Usually they are more easily adapted to automatic dumping mechanisms and can even be arranged to dump without stopping as the car passes the hopper. They lose their advantage where it is necessary to move the equipment in more than one direction, as on a circular pit track with a switch-back to the dump, for to do so the cars must be turned around or arrangements made for dumping on either side of the track.

A surplus of car equipment over maximum needs provides means of reducing time loss due to repairs. Any unit needing repairs can be shunted to the repair track and replaced by another. Thus, the haulage system is not held up by repairs to each unit. This has the effect of giving preference to more small cars than a few large ones.

The power required to move a car or train of cars is equal to the resistance offered by the car or train. The resistance offered by a train is that required (1) to overcome the inertia of standing equipment, (2) to accelerate from one speed to another, and (3) to maintain any attained speed. The resistance offered by inertia will range from 1 1/2 to 5 or more times the running resistance. It will be greater in cold weather, on dirty track, and after prolonged standing of cars.

The resistance offered by the inertia of a standing train is usually omitted in computing the locomotive power for gravel-pit practice, because the inertia of the train is seldom exerted as a single unit. Instead, the inertia of each car is overcome separately. There is always a certain amount of slack in the couplings between cars. In common practice the locomotive is first reversed and the cars are pushed together. When the locomotive then moves ahead, it starts the first car as a single unit. The momentum of the first car helps to start the second and so on as the slack is taken up in each coupling. In this way, train inertia is overcome a single car at a time. If the cars in a train were coupled tightly with no slack in the couplings, the locomotive usually employed would seldom be able to start a loaded train. For this reason, in practice it is easier to start a given pay load in a number of small cars than the same load concentrated in fewer large cars. This practice cannot be used with cars having no brakes if they must be started on an adverse grade.

The theoretical resistance due to acceleration can be computed from the following formula:

$$P = 70.22 \frac{(V^2 - v^2)}{S}$$

S

in which

P = resistance per ton of gross train weight, in pounds;
 S = distance between two points at which velocities are v and V, in feet;
 v = initial velocity (for a train at rest v = 0), in miles per hour; and
 V = final velocity, in miles per hour.

The time consumed in accelerating to the required speed in the designated distance may be computed from the following formula:

$$P = \frac{95.6 V}{t} \text{ or } t = \frac{95.6 V}{P}$$

Table 42 has been computed from these formulas to show the accelerative force P necessary to impart various velocities within various distances to 1 ton of gross load exclusive of frictional or other resistances and the time t required, in seconds.

TABLE 42.- Accelerative force, pounds per ton (P), and time (t), seconds

Distance, feet	Velocity, miles per hour																			
	3		4		5		6		7		8		9		10		15		20	
	P	t	P	t	P	t	P	t	P	t	P	t	P	t	P	t	P	t	P	t
100.....	6.3	45	11.2	34	17.5	27	25.3	23	34.4	20	45.0	17	56.7	15	70.0	14	158	9	281	7
150.....	4.2	68	7.5	51	11.7	41	16.8	34	23.0	29	30.0	26	38.0	23	47.0	20	105	14	187	10
200.....	3.2	90	5.6	68	8.8	54	12.6	46	17.2	39	22.5	34	28.5	30	35.0	27	79.0	18	140	14
250.....	2.5	115	4.5	85	7.0	68	10.1	57	13.8	49	18.0	43	22.7	38	28.0	34	63.0	23	112	17
300.....	2.1	137	3.7	103	5.8	82	8.4	68	11.5	58	15.0	51	19.0	45	23.4	41	52.5	27	93.5	20
400.....	1.6	180	2.8	137	4.4	109	6.3	91	8.6	78	11.2	68	14.2	61	17.5	55	39.5	36	70.0	27
500.....	1.3	220	2.2	174	3.5	137	5.0	115	6.9	97	9.0	85	11.3	76	14.0	68	31.5	46	56.3	34
600.....	1.1	260	1.9	200	2.9	165	4.2	137	5.7	117	7.5	102	9.5	91	11.7	82	26.4	54	46.7	41
700.....	.9	320	1.6	240	2.5	191	3.6	160	4.9	137	6.4	120	8.1	106	10.0	96	22.5	64	40.0	48
800.....	.8	360	1.4	273	2.2	218	3.2	180	4.3	156	5.6	137	7.1	121	8.8	109	19.5	74	35.0	55
900.....	.7	410	1.2	320	1.9	252	2.8	205	3.8	176	5.0	153	6.3	137	7.8	123	17.5	82	31.2	61
1,000.....	.6	478	1.1	348	1.8	266	2.5	230	3.4	197	4.5	170	5.7	151	7.0	137	15.8	91	28.1	68
1,500.....	.4	720	.7	510	1.2	400	1.7	338	2.3	291	3.0	255	3.8	227	4.7	203	10.5	137	18.7	102
2,000.....	.3	960	.6	638	.9	530	1.3	440	1.7	394	2.3	333	2.9	297	3.5	273	7.9	182	14.0	137

Example 5. What accelerative force will be required to bring a standing train of 10 cars to a speed of 10 miles per hour in a distance of 1,000 feet if each car weighs 5 tons and carries 12 tons of payload?

Total train weight.....	tons	170
Accelerative force per ton for 10 miles per hour and 1,000 feet, from table 42.....	pounds	7.0
Total force required.....	do.	1,190.0

Example 6. What accelerative force will be required to bring the train of example 5 running at 10 miles per hour to a speed of 15 miles per hour in 500 feet? From table 42:

		<u>Pounds</u>
Force for 15 miles per hour	500 feet	31.5
Force for 10 miles per hour	500 feet	<u>14.0</u>
Increased force per ton.....		17.5

Total force equals $170 \times 17.5 = 2,975$ pounds.

This means that in addition to the drawbar pull necessary to move the train the locomotive would have to have an extra pull of 1,190 pounds to attain a speed of 10 miles per hour in 1,000 feet and 2,975 pounds to accelerate to 15 miles per hour in the next 500 feet.

The resistance offered by a running train is the cumulative effect of a number of factors, such as (1) wind resistance, (2) lubrication, (3) oscillation and concussion between wheels and rails, (4) rolling friction between wheels and rails, (5) journal friction between axles and boxes, (6) track grade, and (7) track curvature.

In sand and gravel pit operation the first five factors are impossible of determination with mathematical precision. This is true because of the multiplicity of car, wheel, and journal design, rail used, and track lay-out. Generally wind resistance will be negligible because of the comparatively low speeds. Lubrication will range from poor to medium, with few conditions comparable to good railway operation. Oscillation and concussion will vary with the square of the velocity. Their determination independent of other resistances cannot be computed, but because of the necessarily poor alinement and the condition of pit tracks they will be at a maximum. Rolling friction has not been definitely determined independent of other friction factors. It depends upon the rigidity of the rail and the support given by the ties. For the same reason this also will be high. Journal friction per ton of load is less for higher pressures or heavy wheel loads, greater for very low or high speeds, a minimum for train speeds of about 10 miles per hour, and less in warm weather. Obviously, all local conditions must be known before calculations can be made.

These factors are more definite and can be fairly closely evaluated after experiment and trial for standard railway equipment and conditions, and some laboratory experiments have been made on coal-mine equipment.³

Various formulas have been compiled by different authorities from which to compute the resistance of a train due to its speed, but none can be used accurately under the varied conditions of sand and gravel operations. Train resistance over a particular section of track or at a particular point is determined most accurately by the use of a dynamometer car connected between the locomotive and the train. This car is equipped with electrical instruments which make a graphic record of the exact total force used in moving the train. By the use of such devices experiments have shown that pit haulage on level track requires a drawbar pull ranging from 10 to as high as 60 pounds per ton of gross train weight. The minimum of 10 pounds may be said to apply where cars in good condition are operated over straight, well-ballasted tracks. Except under the worst car and track conditions, it is doubtful if more than 40 pounds per ton will be required. The same experiments also show that there is little difference in train resistance when running at 5 to 10 miles per hour, although at lower velocities the resistance increases. In common practice, pit haulage in sand and gravel operations seldom will average over 10 miles per hour, although in some instances higher speeds are used for short distances to build up momentum to carry the train over short, steep grades.

Train resistance for gravel-pit haulage may be taken as ranging from 10 to 40 pounds per ton of gross weight, but it may increase to 60 pounds under particularly adverse conditions.

³ Hersey, M. D., and Wetzel, H. E., Mine-Car Friction as Influenced by Wheel Diameter and Other Variables: Carnegie Inst. Technol. Bull. 13, 1924, pp. 35.

Hersey, M. D., Shove, H., and Downes, M. S., Mine-Car Friction with Six Types of Trucks: Carnegie Inst. Technol., Bull. 20, 1925, pp. 33.

Example 7. The 10-car train used in example 5 might offer a running resistance as low as

$$10 \times 170 = 1,700 \text{ pounds,}$$

or might under extreme conditions require

$$60 \times 170 = 10,200 \text{ pounds,}$$

but ordinarily it would not exceed

$$40 \times 170 = 6,800 \text{ pounds.}$$

Train resistance due to grades can be computed accurately, since it varies only with the grade. Track grades are expressed in various ways, such as percentage, feet rise in 100 feet horizontal advance, degree (degree of angle above horizontal), and feet per mile. Table 43 shows the equivalents of these methods of expression.

TABLE 43.— Grade equivalents

Percent	Feet per 100 feet	Degrees		Feet per mile
1/4.....	0.25	0°	9'	13.2
1/2.....	.50	0°	17'	26.4
3/4.....	.75	0°	26'	39.6
1.....	1.00	0°	35'	52.8
2.....	2.0	1°	10'	105.6
2 1/2....	2.5	1°	30'	132.0
3.....	3.0	1°	45'	158.4
3 1/2....	3.5	2°	0'	184.8
4.....	4.0	2°	20'	211.2
5.....	5.0	3°	0'	264.0
6.....	6.0	3°	20'	316.8
7.....	7.0	4°	0'	369.6
8.....	8.0	4°	45'	422.4
10.....	10.0	5°	45'	528.0
12 1/4..	12.25	7°	0'	647.0
14.....	14.0	8°	0'	740.0
16.....	16.0	9°	0'	845.0
18.....	18.0	10°	6'	950.0
20.....	20.0	11°	20'	1,056.0
25.....	25.0	14°	0'	1,320.0

The formula to be used in determining the resistance due to grade will depend upon how the grade is expressed.

Percent.....	$R = 20 n$
Feet per 100 feet	$R = 20 F$
Degree.....	$R = W \text{ sine angle}$
Feet per mile.....	$R = 0.379 H$

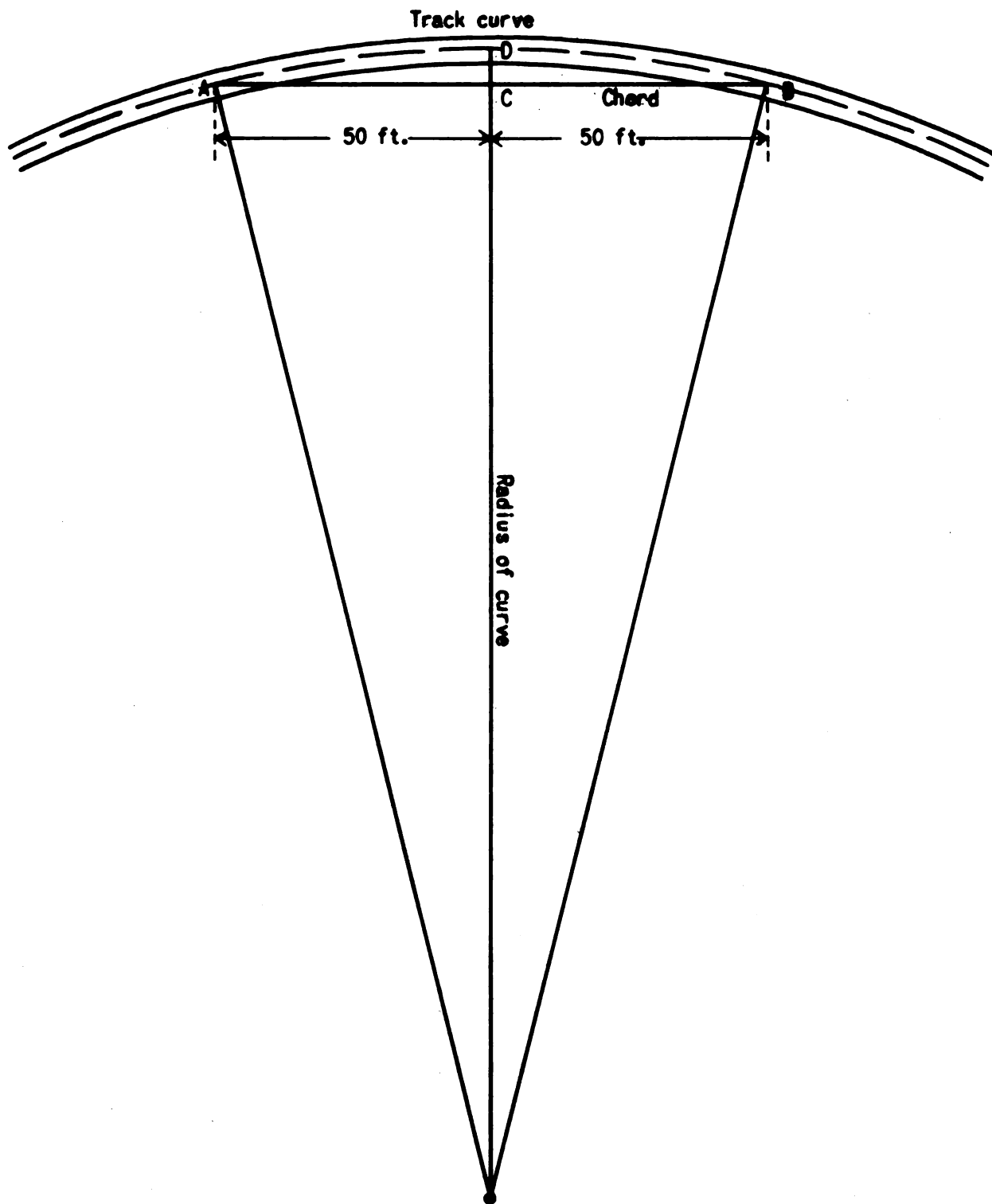


Figure 15.- Method of curve measurement.

In these formulas

R = pounds of resistance per ton of gross train weight,
 n = number of percent,
 F = feet rise in each 100 feet advance,
 W = 2,000 pounds of gross train weight,
 Sine angle = natural sine of the angle of inclination or
 height rise in feet divided by the inclined length in feet, and
 H = height of rise in each horizontal mile.

Calculation shows that up to a 10-percent grade the resistance is exactly 20 pounds per ton of gross train weight for each percent of grade. Thus, the resistance to 1 ton of gross load on a 1/4-percent grade will be $20 \times 1/4 = 5$ pounds; on a 5-percent grade it will be $20 \times 5 = 100$ pounds; etc.

The resistance due to track curvature depends upon the friction of the wheel flange on the outer rail, the compensating elevation of the outer rail, the length of the wheel base of the car, whether the wheels are solid or loose on the axles, whether the trucks are free to turn or rigidly attached to the car, the velocity of the car, and the track gage.

Because of these factors curve resistance cannot be reduced to mathematical precision but must be ascertained experimentally, as with the dynamometer car. Published results of experimental data are scarce. University of Illinois Engineering Experiment Station Bulletin 167 shows curve resistances ranging from 0.41 to 0.64 pound for freight trains on 1° and 3° curves or an average of 0.5 pound per ton per degree of curve. Another authority (anonymous) showed 21 pounds friction per ton for a single car due to a curve of 85-foot radius, or 0.3 pound per ton per degree of curve. He also found a 20-car train on a curve of 350-foot radius gave 7 pounds per ton or 0.4 pound per ton per degree of curve. Other investigators find curve resistances ranging from 0.5 pound to 1.72 pounds per ton per degree of curve. Authorities are inclined to agree that, for permanent track, curve resistance should be computed as ranging from 0.5 to 0.8 pound per ton per degree of curve. For temporary or movable track such as that found in gravel pits the resistance will be higher, and probably not less than 1 pound per ton per degree of curve should be used.

Railway curves are designated by degree of curve or radius of curve. The radius of the curve is the distance from the center line between the rails to the center of the circle of which the curve is an arc. The degree of the curve is the number of degrees in the angle subtended by a chord 100 feet long measured in a straight line between two points on the curve.

Figure 15 shows a railway curve with the parts mentioned indicated. The curve shown is a 28° curve, as measured by the angle subtended by the 100-foot chord, and has a radius of 206 feet. A convenient way of measuring the radius of a curve without locating the center of the circle is as follows:

As in figure 15, measure the distances AC and CD in inches. Multiply each by itself and add together. Divide this sum by twice the length CD in inches. This will give the radius of the curve in inches. The example shown in figure 15 will work out as follows:

AC = 50 feet = 600 inches;	$600 \times 600 = 360,000$
CD = 6 feet 2 inches = 74 inches;	$74 \times 74 = \underline{5,476}$
Add =	365,476
Twice CD = 148 inches.	
$365,476 \div 148 = 2,470$ inches or a radius of 206 feet.	

Care must be exercised to see that the line AB, as laid out, is straight and that CD is measured accurately, as an error of a fraction of an inch here may make a difference of as much as 1°.

Distance AB may be measured on the inner rail rather than on the center line of the track and may be any convenient distance down to, say, 20 feet for sharp curves. Measurements should be made from and to the same part of the rail in each case.

For approximate calculations the measurements can be made in feet using the following formula:

$$\text{Radius} = \frac{(AC)^2 + (CD)^2}{2 \text{ CD}}$$

Table 44 has been compiled to show the degree of curvature corresponding to the length of radius.

TABLE 44.- Degree and radius (feet) of curves

Curvature, °	Radius	Curvature, °	Radius	Curvature, °	Radius
1	5,730	21	273	41	139
2	2,866	22	260	42	136
3	1,910	23	249	43	133
4	1,432	24	238	44	130
5	1,146	25	229	45	127
6	955	26	220	46	125
7	818	27	212	47	122
8	716	28	206	48	119
9	636	29	197	49	117
10	573	30	191	50	114
11	521	31	185	51	112
12	477	32	179	52	110
13	441	33	174	53	108
14	409	34	169	54	106
15	382	35	163	55	104
16	358	36	159	56	102
17	337	37	155	57	100
18	318	38	150	58	98.9
19	301	39	147	59	97
20	286	40	143	60	95.5

The total resistance offered by a car or train is, then, the sum of the resistance due to (1) acceleration, (2) rolling and journal friction, (3) grade, and (4) curvature.

Example 8. What is the total resistance offered by a train of 10 gravel cars each weighing 5 tons and carrying 12 tons of pay load running at 10 miles per hour on a 2-percent grade and a 3° curve? It is assumed that cars and track are in average condition.

The gross weight of the train is 10 x 17 = 170 tons.

	Pounds
Rolling resistance, 170 x 30.....	5,100
Resistance due to grade, 170 x 20 x 2.....	6,800
Resistance due to curve, 170 x 3 x 1.....	510
Total.....	12,410

If the train must reach this speed in a distance of 2,000 feet, resistance due to acceleration (table 42) is then

$$170 \times 3.5 = 595 \text{ pounds.}$$

If the train starts and continues the whole 2,000 feet on the grade and curve above, the total resistance will be $12,410 + 595 = 13,005$ pounds.

The starting resistance of the train as a unit at the instant of starting under good conditions may be

$$13,000 \times 1 \frac{1}{2} = 19,500 \text{ pounds,}$$

but under adverse conditions it may be as much as

$$13,000 \times 5 = 65,000 \text{ pounds.}$$

In other words, this is the drawbar pull necessary to be exerted by the locomotive if the train is started as a unit and is exclusive of the force necessary to move the locomotive itself. However, as stated before, the starting resistance is usually omitted in computing drawbar pull for pit locomotives.

The speed available from the motive power is usually of secondary importance, because it is seldom that pit tracks are in condition to permit the maximum speed of the locomotive. On permanent tracks for long hauls, however, conditions are different, and the speed of the equipment becomes important in reducing the haulage cycle.

LOCOMOTIVE POWER

The motive power and speed of the locomotive or hoist probably has more effect on the capacity of a haulage system than any other element.

Locomotive design ranges from the machine which has comparatively low drawbar pull but which easily maintains high speeds to the heavy, slow-speed engine capable of starting heavy loads.

Details of design vary with the type of power or fuel used, gage of track, and nature of the work to be done. Generally, long, level hauls use lighter locomotives capable of higher speeds.

Ordinarily in sand and gravel pits locomotive speed is sacrificed for power. In fact, in many installations the locomotive is provided with a governor by which the speed is held to a predetermined maximum.

The function of a locomotive is to provide power for moving. To do this with greatest efficiency it must be in motion the greatest possible percentage of operating time. Thus, delays of any nature either at the digger, dump, or enroute impair locomotive efficiency.

Proper selection of a locomotive requires intensive study of all local conditions, with particular attention to the length of haul; gross load per trip; and condition of the track as to roadbed, curves, and grades. In addition, careful analysis should be made of the costs of various kinds of fuel or power.

Consideration of length of haul and gross load per trip involves study of the probable time cycle and therefore the speed required. It also involves study of the weight and tractive effort necessary to handle the required load.

Consideration of the resistance offered by the roadbed, curves, and grades also involves a study of speed and tractive effort, but the emphasis in this case is on the latter.

A locomotive capable of handling the calculated gross load per trip in the required time cycle over a level track might be entirely inadequate were it necessary to put in a slight adverse grade or a sharp curve.

As a general rule it is better to select a locomotive that will provide ample capacity under maximum adverse conditions. The cost of extra power over and above that needed for favorable conditions can easily be exceeded by that of delays due to inadequate power under adverse conditions. Just how much surplus power an operator is justified in providing is largely a matter of individual opinion and may range from 25 to 100 percent of normal requirements under favorable conditions. Ordinarily, however, if the maximum possible adverse conditions are calculated for each factor and the selection is made on this basis the unit should be satisfactory. Allowance for future increased production should be made by additional units rather than excess size at the start.

The selection of the type of power to be used also depends upon local conditions. Coal-fired steam locomotives have in the past proved most popular. However, advances made in the design and flexibility of the mechanical application of internal-combustion engines have caused many operators to show preference for the gasoline or Diesel locomotive.

Electric power so far has found little favor with sand and gravel producers.

Steam power is flexible and permits the slow movement necessary in starting and accelerating a load as well as higher speed when under way. Steam locomotives are capable of considerable overload when adverse conditions require excess power. However, they require periodic supplies of fuel and water, and usually, but not always, some working time is lost in supplying them.

Large steam locomotives require two men to operate them, an engineer and a fireman, either or both of whom may be required to pass certain State examinations for qualification. In smaller locomotives the engineer often does his own firing. Steam locomotives require time to warm them up and generate steam before they are ready to operate.

Gasoline or Diesel locomotives are usually designed to provide maximum power when operating at a certain engine speed. Flexibility in running speed has been provided by including two or more gear ratios at which the locomotive speed can be altered but keeping the engine speed the same. They are capable of an appreciable overload. They also require periodic supplies of fuel, but ordinarily they are supplied with fuel tanks sufficient to carry them through a working period. One man only is needed to operate them, and usually no State regulatory laws as to qualification apply. In cold weather gasoline or Diesel motors are sometimes hard to start, but this is usually eliminated by providing housing facilities for them when off duty.

In the larger sizes combination Diesel-electric locomotives have been demonstrating considerable economy in railway and switching operations according to some users. In this type a Diesel engine is used to drive an electric generator which furnishes power for the driving motors; all are on the same chassis. This combination, however, is usually applied to long hauls with standard equipment rather than to gravel-pit haulage.

The application of outside electric power to locomotives requires either an overhead trolley or a third rail to carry the current, or the locomotive must be equipped with a large storage battery.

Overhead-trolley and third rail systems have been frowned upon by sand and gravel operators for various reasons. Some dislike them because of the danger to workmen. Others object because of the cost of maintaining the bond between rails, the expense of carrying the trolley line along with the track, and the additional cost involved in shifting electrical apparatus along with the pit track. The principal objection to the storage-battery type has been its first cost and slow speed.

All these objections are founded on the experience of gravel-pit operators. However, producers of other materials have found both types of electric locomotives flexible and

economical. Manufacturers of the trolley or third-rail type have equipped their machines with automatic reels which carry two armored cables. By the use of these reels the trolley or third rail can be terminated at the end of permanent track. At this point the ends of the cables can be attached to the rail and trolley, and the locomotive can then proceed on power supplied from the cable. The reels are automatic in that as the locomotive is reversed they rewind the cable previously paid out. Cable length is limited in practice to about 1,000 feet.

Another arrangement is provided in which a wire rope is carried on a reel at the front of the locomotive. This can be stretched by hand to reach cars beyond the reach of the power line. By this means the locomotive can pull in such outlying loads.

The storage-battery locomotive requires considerable time for recharging. Suitable equipment for recharging must also be supplied. Once charged, however, it is just as mobile as either the steam- or oil-powered machine. If the battery charge is reduced to the danger point some types can be recharged through the trolley wire; others must be returned to a central charging station. If the battery charge is depleted below that necessary to handle the train it will usually still retain enough stored energy to return the locomotive, although the train may have to be abandoned.

The electric locomotive can be used for either low or high speeds and is capable of overloading for short periods.

It requires no fuel supply other than lubricants but must have power lines or facilities for battery charging. In some instances batteries are replaceable; as soon as one is depleted it is removed and placed on the charging rack and a freshly charged battery is placed on the locomotive chassis.

Electric locomotives are operated by one man, and no regulatory examinations are required.

In addition to these types of locomotives there are others less often used but worthy of mention.

The compressed-air locomotive is essentially an air receiver on a locomotive chassis. The receiver is charged with compressed air under high pressure and uses this expansively in suitable driving engines. Like the storage-battery type, it must be returned to a charging station for a fresh supply of power.

A somewhat similar type is the fireless steam locomotive. The receiver in this case is partly filled with water and is then charged with steam under high pressure at a charging station. The expanding steam is utilized in the same way as in a locomotive making its own steam. This type also must be returned to the charging station for a fresh supply of power.

Electric, compressed-air, and fireless steam locomotives are particularly advantageous for use underground or wherever the gases of combustion from coal- or oil-fired steam locomotives or the exhaust gases from gasoline or Diesel engines are detrimental to health.

Electric power through a third rail has been successfully applied to a method of operating single cars equipped with their own motive power and operating without an accompanying attendant. These systems have been designed to use direct or alternating current, and by suitable arrangements only a portion of the track is electrified at any one time. In operation the cars are controlled by one man stationed in a centrally located tower from which he can see all cars at all times. By means of a properly constructed control panel he can start, stop, or back a car at any point on its circuit. The system is expensive in first cost but is said to be low in operating cost. Fewer but more expensive cars are required, and users claim that the delay at the digger due to lack of empty cars is a minimum. The system is open to the same objection as that of the trolley locomotive where tracks must be shifted at frequent intervals. Where the pit affords a long circular or elliptical working face in which the track can parallel the route of the digger and the volume of material warrants the initial expense, these remote-control systems offer interesting possibilities.

Condition of Track

Gravel-pit track may be permanent or temporary, depending on the design of the haulage system. Where a long straight or slightly curved working face is available and the track parallels the path of the excavation unit, it pays to ballast the track and build it in accordance with at least semipermanent design. Where it is necessary to shift the track with each move of the excavator this is impossible, and the track must be temporary. Usually, at some distance from the excavator (varying with working conditions), the track can be made permanent and well-ballasted to and from the dump.

Occasionally the difference in elevation between working face and dump is too great for a single grade, and no space is available for spiral ascent. In such cases switch-backs may be employed, or the locomotive may be replaced or supplemented by a hoist operating on an inclined plane.

Track gage ranges from a minimum of 20 inches to a maximum of 4 feet 8 1/2 inches or standard railway gage. In general, no rule can be laid down as to the relation between track gage and car capacity or locomotive weight, although one manufacturer definitely limits cars of 6 cubic yards or over to standard gage. Ordinarily, however, 24-inch gage is the minimum used with locomotive haul. Judged by number of installations, 36- and 42-inch gages are most popular in the so-called narrow-gage field. Generally, the maximum weight for locomotives on 24-inch gage track is 8 to 10 tons and on 30 to 36 inch gage, 12 to 25 tons; over 25 tons requires standard gage. The same may be said to apply to the gross weight of cars.

Narrow-gage track is usually laid with lighter rails and therefore is more easily and quickly shifted. It can be purchased in standard lengths with steel ties attached, which facilitates relaying when frequent shifting is necessary. Locomotive speed is usually lower with narrow-gage track, and such track can therefore be laid with less care as to curvature and grades.

Standard gage permits railway equipment to be shunted to any part of the pit but requires heavier rails, easier curves, lower grades, and more initial expenditure if railway equipment is used. Standard gage permits larger locomotives and cars and consequent lower per-ton-mile haulage cost for labor power and repairs. It is not justified for pits with small tonnage outputs, although just where the line should be drawn between small and large tonnage depends upon local conditions.

On sharp curves the gage must be widened to reduce friction. For standard gage on curves of over 8° the gage should be widened 1/8 inch for each 2° of curve to a maximum of 4 feet 9 1/4 inches.

The proper ratio of rail weight to locomotive weight involves the distribution of the locomotive weight on each drive wheel and the spacing of the ties under the rail. Pit locomotives are usually of the 4- or 6-wheel drive type, and tie spacing usually averages 24 inches center to center. At some plants ties may be spaced as close as 18 inches, while at others they may be spaced 36 inches or more in portable track. Table 45 offers suggested safe combinations of tie spacing, weight of rail, and weight on locomotive drivers.

Sharp curves in the track should be avoided if possible because of the increased resistance to movement of equipment. In pit tracks, particularly those of a temporary nature, this is not always possible. Where necessary, curves should never be sharper than required by the length of the locomotive wheel base. Double-truck pit cars are usually so constructed that each truck has a limited rotative movement which permits the use of sharper curves than their total wheel base would require if rigid.

TABLE 45.- Relation between tie spacing and rail and locomotive weight

Locomotive weight, tons	4		6		8		10		13		15		20	
Number of drivers	4	6	4	6	4	6	4	6	4	6	4	6	4	6
Wheel load, pounds	2,000	1,333	3,000	2,000	4,000	2,667	5,000	3,333	6,500	4,333	7,500	5,000	10,000	6,667

Ties Spaced at 18 Inches

Rail weight, pounds per yard	16	16	20	16	25	20	30	25	40	30	40	30	50	40
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Ties Spaced at 24 Inches

Rail weight, pounds per yard	20	16	25	20	30	25	40	30	50	40	50	40	55	50
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Ties Spaced at 36 Inches

Rail weight, pounds per yard	25	20	30	25	40	30	40	30	50	40	55	50	65	55
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Curves laid out on permanent track should have the outer rail raised somewhat to facilitate the movement of cars. The amount of elevation depends upon the speed at which cars travel around the curve and the degree of the curve. A common rule for determining the elevation of the outer rail is 1 inch for each degree of curve for level track and, if the grade is over 2 percent, 3/4 inch for each degree of curve. This corresponds to a train speed of 38 miles per hour. Table 46 gives the elevations for lower speeds.

TABLE 46.- Elevation of outer rail, inches

Curve, °	Speed, miles per hour				
	15	20	25	30	35
1.....	1/8	1/4	3/8	5/8	7/8
2.....	1/4	1/2	7/8	1 1/4	1 3/4
3.....	1/2	7/8	1 1/4	1 7/8	2 1/2
4.....	5/8	1 1/8	1 3/4	2 1/2	3 3/8
5.....	3/4	1 3/8	2 1/8	3 1/8	4 1/4
6.....	7/8	1 5/8	2 5/8	3 3/4	5 1/8
7.....	1 1/8	1 7/8	3	4 3/8	5 7/8
8.....	1 1/4	2 1/4	3 1/2	5	6 3/4
9.....	1 3/8	2 1/2	3 7/8	5 5/8	7 5/8
10.....	1 1/2	2 3/4	4 1/4	6 1/4	8 1/2
11.....	1 3/4	3	4 3/4	6 7/8	
12.....	1 7/8	3 1/4	5 1/8	7 1/2	
13.....	2	3 5/8	5 5/8	8 1/8	
14.....	2 1/8	3 7/8	6	8 3/4	
15.....	2 3/8	4 1/8	6 1/2	9 1/4	

The horsepower of the power unit on any locomotive can be calculated from its construction characteristics in the same way as that for a similar unit in any other service. This, however, is only an indirect measure of the power of the locomotive. The power unit uses its energy to revolve the drive wheels of the locomotive and through such revolution to move the locomotive and any load to which it is attached. In converting the revolving motion of the drive wheels to linear motion of the locomotive other factors are involved.

Let us assume that a locomotive is placed on the end of a track with its front end against an immovable block so that it cannot move forward. Now, if the power unit is started, the drive wheels cannot revolve until enough force is applied to overcome the adhesion between the tires and rails. If sufficient force is applied, the drivers will revolve or slip on the rails, but the locomotive cannot move.

Now assume the same locomotive is placed on a straight, level track without anything to prevent its movement, and sufficient grease or other lubricant is placed between the rails and drivers to prevent any adhesion. When power is applied through the power unit the drivers revolve, but again the locomotive does not move. In both cases the power unit functions, but the locomotive does not. In the first case the power applied was a maximum and in the second case a minimum.

It is evident that for the locomotive to function there must be adhesion between tire and rail and that the applied power cannot exceed that which the adhesion will support, as otherwise the wheels will slip. The adhesion between tire and rail depends upon the material of which the tire and rail are made, the weight on the tire, and the condition of the surfaces in contact. Locomotive wheels are made of cast iron or cast steel, and they may or may not have forged-steel tires. Rails are made of steel. The adhesion is greater between a steel tire and a steel rail than between a cast-iron tire and a steel rail. The adhesion between tire and rail varies directly with the weight on the tire.

Adhesion varies also with the condition of the surfaces in contact. Thus, a rail covered with dry snow, sleet, grease, or dew will present a variety of conditions all offering less adhesion than if the rail were clean and either thoroughly wet or dry. There is little difference in a thoroughly wet or dry rail if it is clean. In any of the cases cited, however, the application of sand to the rail will raise the adhesion.

The power of a locomotive is measured by the tractive effort of which it is capable. The tractive effort is calculated by multiplying the weight of the locomotive carried on the drive wheels by the coefficient of adhesion between tire and rail.

The coefficient of adhesion is the tractive force required to slip the drivers divided by the weight on the drivers.

The Baldwin Locomotive Works has determined the coefficient of adhesion for various conditions, as presented in table 47.

TABLE 47.- Approximate coefficient of adhesion

Condition of rails	Chilled cast-iron wheels		Steel-tired wheels	
	Without sand	With sand	Without sand	With sand
Covered with dry snow	0.10	0.15	0.10	0.15
Covered with sleet.....	.15	.20	.15	.20
Greasy, moist.....	.15	.25	.15	.25
Outdoor moist.....	.18	.22	.18	.22
Thoroughly wet, clean	.20	.25	.25	.31
Dry, clean.....	.20	.30	.25	.37
Best conditions.....		.38 to .42		.47 to .52

From the table it is seen that with dry rails the tractive effort of which a locomotive is capable will vary with cast-iron wheels from 20 to 25 percent of the weight on the drivers and with steel tires from 25 to 30 percent. Thus, a locomotive weighing 6 tons will be capable of a tractive effort of 2,400 to 3,600 pounds.

It will be noted that the tractive effort is calculated without reference to the horsepower of the power unit. Power units are usually designed to supply the maximum tractive effort with maximum adhesion at starting speeds.

Another unit used to measure locomotive power is the drawbar pull of which it is capable. This is always less than the tractive effort. The drawbar pull of a locomotive is that force which it is capable of supplying at its drawbar to pull a trailing load. It is the tractive effort minus the resistance to motion of the locomotive. The resistance of the locomotive is calculated in the same manner as train resistance in the preceding chapter, giving the same consideration to grade and curvature of track.

Tractive force as applied to locomotives is the force applied by the power unit to the rim of the drive wheels. If the power unit is capable of supplying enough force to overcome track adhesion and slip the drivers, the tractive force is greater than the tractive effort and vice versa.

The tractive force of a 2-cylinder single-expansion steam locomotive may be calculated from the following equation:

$$T = 4 \times \frac{\pi d^2 P S}{4 \pi D} = \frac{\pi d^2 P S}{D}$$

in which

T = tractive force, in pounds;
 d = diameter of the cylinder, in inches;
 P = mean effective pressure (85 percent of the boiler pressure);
 S = length of stroke, in inches;
 D = diameter of drive wheel, in inches.

Note that for each revolution of the drive wheel there are 4 pressure impulses from the 2 steam cylinders.

The indicated horsepower of a steam engine may be calculated from the following equation:

$$I. \text{ hp.} = \frac{PLAN}{33,000}$$

in which

I.hp. = indicated horsepower for each cylinder;
 P = mean effective pressure (85 percent of the boiler pressure);
 L = length of stroke, in feet;
 A = area of cylinder, in square inches, $\frac{\pi d^2}{4}$;
 N = number of single strokes per minute.

The train speed must be known to determine N. This may be found from the following equation:

$$LN = \frac{M \times 88 \times 2S}{\pi D},$$

in which

LN = piston speed, in feet per minute;

M = train speed, in miles per hour;

S = stroke, in inches;

D = diameter of driver, in inches.

Substituting for LN, the equation for a 2-cylinder steam locomotive becomes:

$$I.Hp. = 2x \frac{Px \frac{\pi d^2}{4} \times \frac{M \times 88 \times 2S}{\pi D}}{33,000}$$

$$= \frac{88 P d^2 S M}{33,000 D} = \frac{d^2 P S}{D} \times \frac{M}{375}$$

Since

$$\frac{d^2 P S}{D} = T,$$

then

$$I.Hp. = \frac{TM}{375}$$

and

$$T = \frac{375 I.Hp.}{M}$$

The internal resistance of steam engines varies with the design, lubrication, etc., and may range from 6 to 38 percent of the indicated horsepower. The brake horsepower for locomotives commonly used in sand and gravel haulage may be assumed at 75 percent of the indicated horsepower.

Therefore the formula for brake horsepower is

$$B.Hp. = \frac{3}{4} \times \frac{TM}{375} = \frac{TM}{500}$$

and

$$T = \frac{500 B.Hp.}{M}$$

Gasoline Engines

The horsepower of gasoline engines is calculated from a number of different formulas originating with various technical authorities, no two of which agree exactly. The differences are probably due to different engine designs and variations in the efficiency with which the gasoline vapor is mixed and consumed.

Gasoline-engine horsepower can be calculated from the same equation as that for steam-engine horsepower if proper changes are made to account for the use of gasoline power.

The steam-engine equation was:

$$\text{I.Hp.} = \frac{\text{PLAN}}{33,000}$$

In a gasoline or oil-burning locomotive the engine speed can usually be varied between 500 and 1,000 r.p.m., and the locomotive speed can be varied by shifting the transmission gears which change the ratio of the r.p.m. of the motor to that of the drive axle. Therefore, an equation defining piston speed LN in terms of train speed must include the gear ratio between motor and drive axle and must take into consideration the range of motor speeds. Gear ratios on gasoline locomotives are not standardized but vary with different manufacturers. There is a loss of efficiency in transmitting power from the motor to the drive axle through the transmission gears. This varies with different designs but may be assumed at 20 percent for purposes of computation.

In the 4-cycle gasoline engine, which is the type commonly used for industrial locomotives, each piston makes 4 single strokes in 2 revolutions of the drive shaft, but only 1 of these is an explosion or effective stroke. Therefore, the effective strokes per revolution are one half the number of pistons working.

Let R equal the total ratio and C the number of pistons or cylinders.

Then

$$\begin{aligned} L N &= \frac{M \times 88 \times \frac{C}{2} \times S \times R}{\pi D} \\ &= \frac{M \times 88 \times C \times S \times R}{2 \pi D} \end{aligned}$$

Substituting this value for LN in the original steam-engine equation,

$$\begin{aligned} \text{I.hp.} &= \frac{\text{PA}}{33,000} \times \frac{M \times 88 \times C \times S \times R}{2 \pi D} \\ &= \frac{d^2 \text{ PCSR}}{8 D} \times \frac{M}{375} \end{aligned}$$

The equation for tractive force for a 2-cylinder, single-expansion steam engine was:

$$T = 4 \times \frac{\pi d^2 \text{ PS}}{4 \pi D}$$

As there were 2 cylinders, there were 4 effective strokes per revolution. The equation for the 4-cycle gasoline engine must again account for the ratio between motor and drive axle, the number of cylinders, and the effective strokes per revolution.

This equation then becomes:

$$T = \frac{C}{2} \times \frac{\pi d^2 PSR}{4 \pi D} = \frac{d^2 PCSR}{8 D}.$$

Substituting in the previous equation,

$$I.hp. = \frac{TM}{375}$$

and

$$T = \frac{375 \times I.hp.}{M}$$

These are the same as the corresponding formulas for steam engines.

The efficiency of gasoline engines ranges from 75 to 85 percent, averaging 80 percent of the indicated horsepower, but as previously stated there is an average loss of 20 percent in the transmission gearing. Hence,

$$B.hp. = \frac{4}{5} \times \frac{4}{375} \times \frac{TM}{588} = \frac{TM}{588}, \text{ and } T = \frac{588 B.hp.}{M}$$

The horsepower of a 4-cycle gasoline motor may be computed from the steam formula as follows:

$$I.hp. = \frac{PA}{33,000} \times LN,$$

$$N = \frac{C}{2} \times \text{r.p.m. and } L = \frac{S}{12}$$

S = stroke in inches;

substituting,

$$\begin{aligned} I.hp. &= \frac{PA}{33,000} \times \frac{C}{2} \times \text{r.p.m.} \times \frac{S}{12} \\ &= \frac{PACS}{792,000} \times \text{r.p.m.} \end{aligned}$$

But AS = V = the volumetric displacement of each cylinder in cubic inches. Therefore,

$$I.hp. = \frac{PVC}{792,000} \times \text{r.p.m.}$$

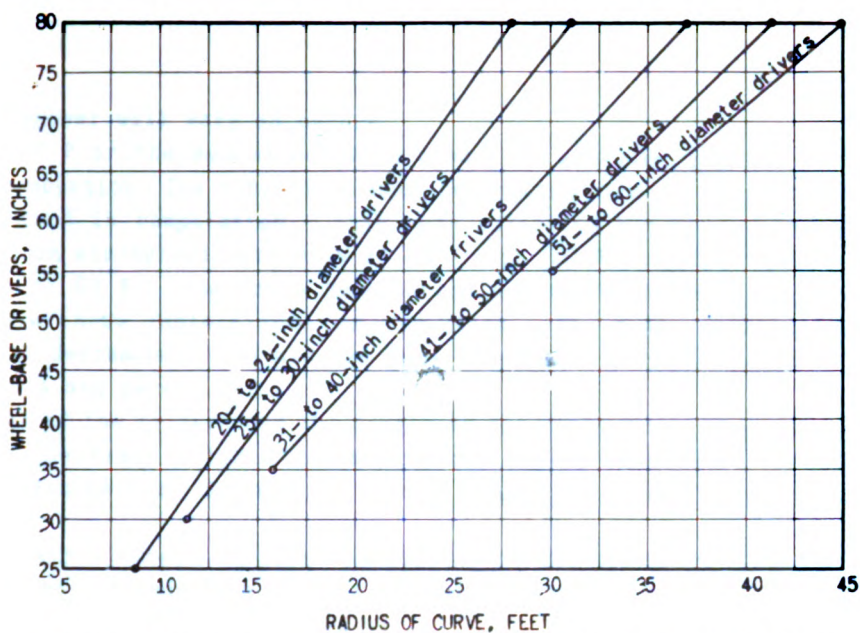


Figure 16.— Relation between diameter of drive wheels, length of wheel base (four-wheel rigid trucks), and radius of curve. Rigid trucks of six or more wheels can negotiate these curves if the center wheels have sufficient lateral clearance or have plain tires without flanges.

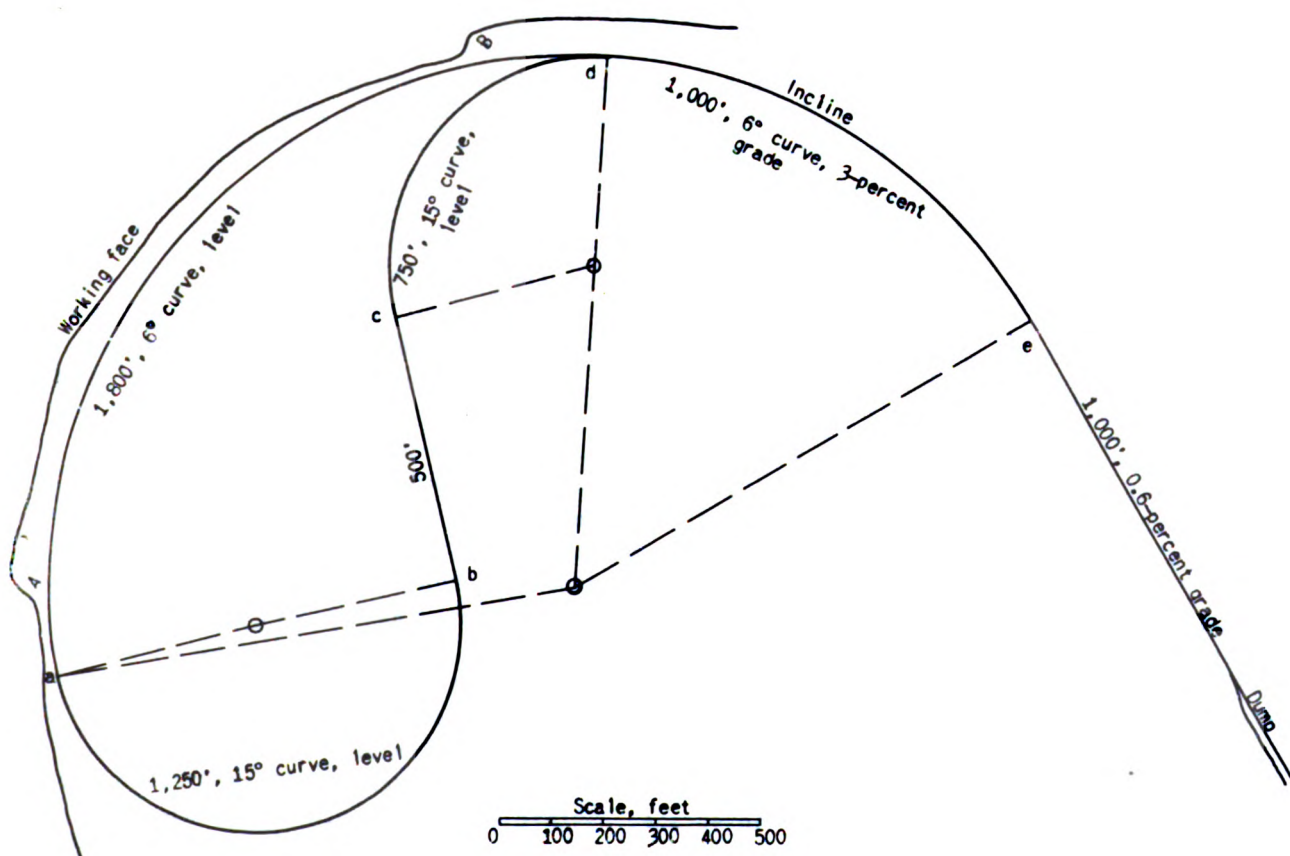


Figure 17.— Pit-track arrangement.

Brake horsepower will vary as before.

The value of P or the mean effective pressure in gasoline engines will vary with the efficiency of combustion. The range in pounds per square inch will be from 65 to 85 with 75 to 80 commonly used in computation. This mean effective pressure as applied to gasoline engines is based on atmospheric pressure at sea level (29.92 inches of mercury) and a standard temperature of 60 F. Accurate computation of the horsepower of internal-combustion engines must take these factors into consideration. The power developed by an internal-combustion engine decreases at an average rate of about 3 percent for each 1,000-foot rise in elevation above sea level.

The diameter of the drivers and the length of the rigid wheel base determines the degree of curvature in the track which a locomotive can safely negotiate. Figure 16 shows the approximate relation between these three items. The curve radius is the minimum that should be used.

Pit haulage with locomotive power is limited by the increased tractive force required for each degree rise in grade to a maximum of 6 percent except for short distances. In fact, customarily grades are kept below 4 percent.

It may be pertinent here to mention one type of locomotive which may be used on steeper grades but which has been used comparatively little in gravel-pit operation. This machine has a pinion on the axle between its drive wheels. On steep grades a rack is laid between the rails and fastened to the ties. As the locomotive approaches the grade its pinion engages the teeth on the rack, obviating the necessity for adhesion between tire and rail. This type of locomotive can negotiate grades up to 10 or even 15 percent.

Occasionally locomotives are required to descend heavy grades with loads and return empties up grade. If there is no sharp curve at the foot of the grade no greater locomotive weight may be required. If there is a sharp curve at the foot of the grade and speed down grade must be decelerated, it may be necessary to install a heavier locomotive than would be required to return the empties up grade. If the cars are not equipped with brakes and the entire deceleration must depend on the locomotive brakes, the locomotive should be of such weight as to provide 20- to 25-percent adhesion on the maximum descending grade.

CAPACITY OF LOCOMOTIVE- AND CAR-HAULAGE SYSTEMS

As previously stated, the capacities of locomotive- and car-haulage systems cannot be tabulated because of the many and divergent factors involved in local conditions.

Methods of calculating requirements are perhaps best illustrated by citing a hypothetical example and making calculations accordingly.

Example 9. It is assumed that a gravel operator has just purchased a dry pit and laid out a haulage route, as shown in figure 17. The deposit has a working face averaging 30 feet in height and extending 1,800 feet from a to d (fig. 17). Entrance to the pit is down a 3-percent grade on a 6° curve for a distance of 1,000 feet (e to d). From the top of the grade (e) to the dump the track is 1,000 feet long and straight but has an adverse grade of 0.6 percent. Pit tracks are level and in accordance with the curves, as shown in figure 17.

There are two 1 1/2-cubic yard shovels located at A and B, each 250 feet from the ends of the working face. A switch is provided at the foot of the 3-percent grade (d, fig. 17).

Running clockwise, the haulage length from shovel A will be 3,550 feet and the return 4,750 feet. From shovel B the haul will be 2,250 feet and the return 6,050 feet.

From the track lay-out, all but that paralleling the working face can be laid as permanent track. Therefore, standard gage is selected. However, very little of the track is straight, hence long trains should be avoided because of curve resistance.

It is assumed that the gravel is loose and free from large boulders or clay and that a minimum production of 300 tons hourly is required for an 8-hour day.

Reference to part I, table 9, in the chapter on power shovels shows that the two 1 1/2-cubic yard shovels will have an hourly capacity ranging from 280 to 530 tons. Under average working conditions these two machines should produce the required tonnage in 6 hours, or they will have an excess capacity of 33 1/3 percent. Moreover, if one shovel is incapacitated for any cause the other can produce the required tonnage by working a short overtime stretch.

Working continuously, each shovel will dig and load an average of 200 tons per hour or 3 1/3 tons per minute, which is the equivalent of 1.48 full dipperloads per minute. The effect of partly loaded dippers was considered in computing shovel capacity, hence need not be considered here.

The shovels, it is assumed, can cut a path 40 feet wide from the bank, which in 1,800 feet of working face will supply upwards of 150,000 tons of gravel, or more than 60 days operation for each cut. The track at the working face, therefore, need not be moved frequently. However, since no loading switches are used at the shovels each train must be partly loaded by each shovel. If 2 trains are used each shovel will necessarily be idle while trains are in transit. With 3 trains, shovels may be kept constantly supplied with cars; but since each train must pass twice over the switch at the foot of the 3-percent grade on each trip, there will be six crossings in each complete haulage cycle. This may or may not be serious.

For efficient operation each car should hold at least three full dipperloads. Since both shovels swing 1 1/2-cubic yard dippers, the minimum car size is established at 4 1/2 cubic yards. However, it has already been decided that standard-gage track will be used. Car builders' catalogs list 6 cubic yards as the minimum-size standard car for standard-gage track. Therefore, the minimum car is selected, holding 6 cubic yards or 9 tons of payload per car. The maximum size probably would not exceed 30 cubic yards or 45 tons.

The shovel capacity which the haulage unit must serve is 400 tons per hour or 1 ton every 9 seconds.

In computing the time per haulage cycle, efficiency requires that the loading time - time the cars are at the shovels - should equal transit and dumping time - time the cars are away from the shovels.

The track arrangement in figure 17 shows a total round-trip distance of 8,300 feet, which at an average train speed of 10 miles per hour would require 9 minutes and 26 seconds.

Since the loading, dumping, and transit time for any train varies with the number of cars in the train, the train size must be assumed before a train time schedule can be computed. This necessitates a cut-and-try method of initial calculation.

Assuming that the unit adequately serves the shovels, that the dumping time is 30 seconds per car irrespective of size, and that the transit time is 10 minutes per train, also irrespective of size, the time cycles for various train units is as shown in table 43.

From table 43, a train of twelve 6-cubic yard or 9-ton cars will require 972 seconds to load at each shovel, assuming 6 cars are loaded at each shovel and that transit and dumping time is 960 seconds. The transit plus dumping time is slightly less than the loading time at each shovel, assuring constant shovel operation.

From the same table, the 30-cubic yard or 45-ton car will be used in 2-car trains, the 12-cubic yard or 13-ton car in 5-car trains, and the 20-cubic yard or 30-ton car in 3-car trains.

Since each shovel dipper carries 2.25 tons the 9-ton car will require 4 dipperloads, the 18-ton car 8 dipperloads, and the 45-ton car 20 dipperloads. The 30-ton car, however, will require 14 loads and consequently will carry 31.5 tons. Obviously the selection of the car size is not limited to those sizes cited. The four sizes, however, show how train units may be balanced. In the following computations only the 6-cubic yard car, assumed to weigh 6 tons empty, and the 30-cubic yard car, assumed to weigh 40 tons empty, are used.

TABLE 48.- Train-time cycles (seconds)

Fayload in car, tons	Number of cars in train	Payload in train, tons	Loading time at each shovel	Transit time	Dumping time	Total cycle time
9	5	45	405	600	150	1,560
9	10	90	810	600	300	2,520
9	11	99	891	600	330	2,712
9	12	108	972	600	360	2,904
9	13	117	1,053	600	390	3,096
45	1	45	405	600	30	1,440
45	2	90	810	600	60	2,280
45	3	135	1,215	600	90	3,120
18	4	72	648	600	120	2,016
18	5	90	810	600	150	2,370
30	3	90	810	600	90	2,310

Table 48 shows that the train of twelve 9-ton cars delivers a payload of 108 tons in a time cycle of 2,904 seconds. The haulage duty required to serve the shovels is 2,400 tons in 6 hours or 21,600 seconds. There will then be $\frac{2,400}{108} = 22.22$ train loads required in 6

hours.

A single train can make $\frac{21,600}{2,904} = 7.44$ round trips in 6 hours.

Therefore, $\frac{22.22}{7.44} = 3$ trains required.

Similarly,

$\frac{2,400}{90} = 26.67$ trainloads required of 30-cubic yard cars.

$\frac{21,600}{2,20} = 9.5$ round trips in 6 hours.

$\frac{26.7}{9.5} = 2.8$ or 3 trains required.

With a combination of 2 shovels and 3 trains the loading time at each shovel must approximate the transit plus dumping time for greatest efficiency in the haulage unit.

With good cars well-maintained and the conditions specified as to curves and roadbed, the rolling resistance probably would not be over 30 pounds per ton of gross train weight with the 6-cubic yard cars and 20 pounds with the 30-cubic yard cars.

The 3-percent grade on a 6° curve will present the maximum tractive-effort requirements, and locomotive weight will be computed on this basis.

The gross trailing load for the 12 small cars will then be $12 \times 15 = 180$ tons and for the 4 large cars, $2 \times 85 = 170$ tons.

The required locomotive weight for gravel-pit haulage can be found from the following formula:

$$20 \text{ WK} = \text{FA} + \text{FW} + 20 \text{ CA} + 20 \text{ GW} + \text{CA} + \text{CW}.$$

in which

W = weight of the locomotive, in tons;
 A = weight of the trailing load, in tons;
 G = grade, in percent;
 F = running resistance, in pounds per ton;
 C = curvature of the track, in degrees;
 K = the coefficient of adhesion, in percent.

Assuming that the locomotives will be equipped with steel-tired drivers and thus permit using a coefficient of friction of 25 percent, and substituting in the formula the conditions applicable to the 3-percent grade:

For 6-yard cars -

$$20W \times 25 = (30 \times 180) + 30W + (20 \times 3 \times 180) + (20 \times 3 \times W) + (6 \times 180) + 6W$$

$$500W - 96W = 17,280$$

$$W = 42.8 \text{ tons.}$$

For 30-yard cars -

$$20W \times 25 = (20 \times 170) + 20W + (20 \times 3 \times 170) + (20 \times 3 \times W) + (6 \times 170) + 6W$$

$$500W - 86W = 14,620$$

$$W = 35.3 \text{ tons.}$$

These locomotive weights may be checked by the following computation of train resistance.

		<u>6-cubic yard cars</u>	<u>30-cubic yard cars</u>
Gross train weight =	180 + 42.8	= 222.8 tons	
	170 + 35.3	=	205.3 tons
Rolling resistance =	222.8 x 30	= 6,684 pounds	
	205.3 x 20	=	4,106 pounds
Grade resistance =	222.8 x 20 x 3 =	13,368 pounds	
	205.3 x 20 x 3 =		12,318 pounds
Curve resistance =	222.8 x 6 x 1 =	1,369 pounds	
	205.3 x 6 x 1 =		<u>1,232 pounds</u>
Total resistance =		21,421 pounds	17,656 pounds

At 25-percent adhesion the locomotive weight must be 4 times the total resistance, or

$$4 \times 21,421 = 85,684 \text{ pounds} = 42.8 \text{ tons.}$$

$$4 \times 17,656 = 70,624 \text{ pounds} = 35.3 \text{ tons.}$$

Selection of locomotives at those weights would require the maximum duty of which they were capable on the 3-percent grade. Since no machine works efficiently if operated continuously at its maximum capacity, the actual selection should be locomotives of greater than the calculated weight. In regular work the locomotive should be used at not more than 75 percent of its tabulated tractive effort.

Therefore the maximum tractive effort of which the chosen locomotives should be capable is:

$$\frac{21,421}{0.75} = 28,560 \text{ pounds, and}$$

$$\frac{17,656}{0.75} = 23,540 \text{ pounds.}$$

At 25-percent adhesion they should weigh 57 and 47 tons, respectively.

In steam locomotives the tractive force is calculated from the following formula:

$$T = \frac{d^2 p S}{D},$$

in which

T = tractive force, in pounds;

d = diameter of the cylinder, in inches;

p = mean effective pressure (this is usually taken as 85 percent of the boiler pressure);

S = stroke, in inches;

D = diameter of the driver, in inches.

Thus, a locomotive having a 19-inch cylinder by 24-inch stroke, with a 46-inch driver, will develop a tractive force of 28,800 pounds at 180 pounds boiler pressure, and a 57-ton locomotive is capable at 25-percent adhesion of 28,500 pounds tractive effort.

Moreover, a locomotive having a 17-inch cylinder by 24-inch stroke with a 44-inch driver will develop a tractive force of 24,100 pounds at 180 pounds boiler pressure, and a 47-ton locomotive is capable at 25-percent adhesion of 23,500 pounds tractive effort.

The probable weights chosen from commercial sizes would be 60- and 50-ton locomotives. Having determined the size of the locomotive, the next step in calculating the haulage capacity of the system is computation of the time cycles.

Computation of time cycles involves calculation of the safe speed at which the trains may move. This is limited by the grade, curvature, and condition of the track. It may also be limited to a specific maximum by governing the locomotive.

The safe speed on curves in miles per hour equals the square root of the radius of the curve in feet. The radius of a 1° curve is 5,730.65 feet, and the radius of any curve is 5,730.65 divided by the number of degrees.

The safe speed on grades depends on the weight of the locomotive and train and the condition of the track.

Since the sharpest curve in figure 17 is 15°, the safe speed will be:

$$\frac{5,730}{15} = 19.54 \text{ miles per hour.}$$

Assuming the trains in either case will be run in a clockwise direction,⁴ the haulage route can be segregated into 10 sections, as follows (fig. 17):

⁴ It is often desirable to reverse the position or direction of locomotives periodically in such service to avoid excessive flange wear on one side and distortion of springs and other parts.

Section	Length, feet	Grade, percent	Curve	Speed limit, m.p.h.
B - d.....	250	0	6°	
d - e.....	1,000	+ 3.0	6°	
e - dump	1,000	+ 0.6	0	
dump - e	1,000	- 0.6	0	
e - d.....	1,000	- 3.0	6°	
d - c.....	750	0	15°	19
c - b.....	500	0	0	
b - a.....	1,250	0	15°	19
a - A.....	250	0	6°	
A - B.....	1,300	0	6°	

Assuming a maximum speed of 20 miles per hour and starting with a loaded train at shovel B, the tractive effort and time required for each section are as shown in table 49.

TABLE 49.- Tractive effort and time required

Section	6-cubic yard cars			30-cubic yard cars		
	Gross train weight, tons	Tractive effort, pounds	Time, seconds	Gross train weight, tons	Tractive effort, pounds	Time, seconds
B - d.....	240	15,360	34	220	11,880	34
d - e.....		23,040	69		18,930	69
e - dump.....		10,080	86		7,040	86
Dumping ¹			360			60
Dump - e.....	132	7,590	64	130	6,180	64
e - d.....		-3,170	41		-4,420	41
d - c.....		5,940	27		4,550	27
c - b.....		3,960	18		2,600	18
b - A.....		5,940	67		4,550	67
A loading ²			972			810
A - B.....	186	11,900	123	175	9,450	123
B loading.....			972			810
Total time.....			2,833			2,209

¹Dumping time assumed at 30 seconds per car.

²Loading time calculated at shovel capacity or 3 1/3 tons per minute or 162 seconds for each 6-cubic yard car and 810 seconds for each 30-cubic yard car. It is assumed that cars are spotted during the swing cycle of the shovel without loss of time.

An illustration of the method of computing figures in table 49 follows:

Section B - d

From table 42 resistance due to acceleration to 10 miles per hour in 250 feet is 28 pounds per ton, and the time required is 34 seconds.

Resistance due to curve is 1 pound per degree of curve, or 6 pounds.
 Track resistance is 30 pounds for small and 20 pounds for large cars.
 Total resistance is $28 + 6 + 30 = 64$ pounds for small cars, and
 $28 + 6 + 20 = 54$ pounds for large cars.

$$240 \times 64 = 15,360 \text{ pounds, and}$$

$$220 \times 54 = 11,880 \text{ pounds.}$$

Section d - e

	<u>Small cars</u>	<u>Large cars</u>
Rolling resistance..... pounds	30	20
Grade resistance..... do.	$20 \times 3 = 60$	60
Curve resistance..... do.	$6 \times 1 = \underline{6}$	<u>6</u>
Total resistance..... do.	96	86

$$96 \times 240 = 23,040 \text{ pounds, and}$$

$$86 \times 220 = 18,930 \text{ pounds.}$$

Since the effort required is within the power of the locomotive, a constant speed of 10 miles per hour is assumed. This is at the rate of 14.6 feet per second.

$$\frac{1,000}{14.6} = 69 \text{ seconds.}$$

$$14.6$$

In section e to dump, it is assumed that the speed of 10 miles per hour is continued for 750 feet, requiring 52 seconds, and deceleration takes place in 250 feet or 34 seconds.

In section dump to e, it is assumed that acceleration to 15 miles per hour requires 400 feet and, from table 42, takes 36 seconds. Six hundred feet at 15 miles per hour requires 28 seconds.

In section e - d, the 3-percent grade provides accelerative force over and above the resistance due to rolling friction and curve. The 15° curve at d limits speed to 19 miles per hour; hence the train will have to descend the grade on the brake. An average of 17 miles per hour is assumed for the section.

In sections d - c, c - b, b - a, and a - A, a speed of 19 miles per hour is assumed but decelerated in the last 400 feet.

In section A - B, acceleration to 10 miles per hour is assumed in 250 feet and 34 seconds. Deceleration is computed in the same distance and time at B. The balance of 800 feet is covered at 10 miles per hour.

The dumping time will, of course, depend upon the dumping facilities, but a half minute per car seems reasonable.

Loading time is the shovel loading rate. It is assumed that 6 small cars or 1 large car will be loaded at A and B.

From the time intervals of table 49, a train time sheet may be compiled, as in table 50. Train A starts to load at shovel B at 0 minutes, 0 seconds.

It is evident from a study of the train-time sheet that with three trains of small or large cars:

1. There will be 5- or 6-minute intervals between crossings at switch d.
2. There need be no delay of shovel equipment.
3. Using the small cars there will be a delay of 1 minute 23 seconds for each train, and using the large cars each train will be delayed 3 minutes 41 seconds. Therefore, the complete cycle with small cars will be 48 minutes 36 seconds and for the large cars 40 minutes 30 seconds.

TABLE 50.- Train time sheet

Station	6-cubic yard cars						30-cubic yard cars					
	Train A		Train B		Train C		Train A		Train B		Train C	
	H.	m.	s.	H.	m.	s.	H.	m.	s.	H.	m.	s.
Ar. B	0	0	0	0	16	12	0	32	24	0	0	0
Lv. B	0	16	12	0	32	24	0	48	36	13	30	
Ar. d	0	16	46	0	32	58	0	49	10	14	4	
Ar. dump	0	19	21	0	35	33	0	51	45	16	39	
Lv. dump	0	25	21	0	41	33	0	57	45	17	39	
Ar. d	0	27	6	0	43	18	0	59	30	19	24	
Ar. A	0	28	58	0	45	10	1	1	22	21	16	
Lv. A	0	45	10	1	1	22	1	17	34	34	46	
Ar. B ¹	0	48	36	1	4	48	1	21	0	40	30	
Lv. B	1	4	48	1	21	0	1	37	12	54	0	
Ar. d	1	5	22	1	21	34	1	37	46	54	34	
Ar. dump	1	7	57	1	24	9	1	40	21	57	9	
Lv. dump	1	13	57	1	30	9	1	46	21	58	9	
Ar. d	1	15	42	1	31	54	1	48	3	59	54	
Ar. A	1	17	34	1	33	46	1	49	58	1	1	
Lv. A	1	33	46	1	49	58	2	6	10	1	15	
Ar. B ¹	1	37	12	1	53	24	2	9	36	1	21	

¹All 6-cubic yard car trains wait 1 min. 23 sec. at shovel B and all 30-cubic yard car trains wait 3 min. 41 sec.

4. Three trains of twelve 6-cubic yard cars will deliver 2,376 tons to the dump in 6 hours, and three trains of two 30-cubic yard cars will deliver 2,430 tons in 6 hours.
5. Shovel-capacity service is assured.
6. Plant-capacity requirements are bettered by 30 percent.

HOIST AND ROPE HAULAGE

Hoists, like locomotives, do not in themselves generate power; both depend upon motors or other prime movers. The function of both is to translate the power of the prime mover into motion of the car or train. The locomotive converts that power into locomotive motion and transmits it to the car or train. The hoist remains stationary and converts power to motion by winding a wire rope on its drum.

Hoists may be employed for various haulage units among which are: (1) The engine plane, (2) the gravity plane, (3) the tail-rope system, and (4) the endless-rope system.

Engine Plane

A simple engine place consists of a single track on a plane or incline over which a load is raised or lowered by means of a single wire rope and hoist. The hoist is placed at the top of the incline and may be used to pull either loaded cars or empties up the incline. Motion down the incline is caused by gravity and is controlled by the hoist brake.

A double track may be used on an engine plane, but in such instances two ropes are used. Both may be attached to the same hoist drum, or each may be wound on a separate drum. The hoist pulls loaded cars up one track while the empties descend on the other. With two

ropes on separate drums the drums are usually locked in balance so that the descending empty cars assist in pulling up the loads.

The simple engine plane requires a single track only. The double-rope plane may use a single track for all but a short distance at the half-way point where switches allow the ascending and descending cars to pass, a three-rail track with passing switches half-way, or a double track throughout the whole incline.

With the engine plane, power is used for movement of cars in one direction and they are moved by gravity in the other direction.

Gravity Plane

The gravity plane is essentially a double-track engine plane in which loaded cars descend the incline and gravity is the only motive power. The hoist in this instance is used merely as a brake mechanism and in fact may have no power attachment. Descending loaded cars develop enough power to pull up the empties.

Tail-Rope System

In general, a tail-rope system operates the same way as an engine plane worked in both directions with two ropes. One rope called the "main rope" hauls the loaded cars in one direction, and the other or "tail rope" hauls the empties in the opposite direction. Tail-rope systems may be operated by two hoists, one placed at either end of the haulage route, or they may be worked with a double-drum hoist at one end. In the first instance, the hoist at the dump winds in the main rope and pulls out the tail rope from the other hoist. Both ropes are the same length. In the second instance, the main rope is wound on one drum and the tail rope on the other. The main rope is only as long as the haulage route. The tail rope, however, is twice the length of the haulage route, since it must pass from the hoist drum to a tail sheave at the extreme end of the haulage route and back to the hoist.

Tail-rope systems, like engine or gravity planes, may be used on steep inclines, or they may be used on slight grades or level tracks, since gravity is unnecessary as the motive power in either direction.

Endless-Rope System

As its name implies, the rope used in this system is endless. The rope is moved by a single wheel or drum, and friction is obtained either by a grip wheel or by two or more laps around the wheel. The rope is kept taut by means of tension blocks. In operation the rope travels in one direction at all times, and cars are attached to it by means of friction grips. The cars are attached or detached without stopping rope or hoist.

Endless-rope haulage systems can be used on inclined or level tracks.

Any type of rope haulage may be used to supplement other haulage units. For example, loaded cars may be collected at the excavator and hauled by locomotives to the foot or top of the plane or incline and there attached to the hoist rope.

Under favorable conditions all but the gravity plane may entirely supersede other haulage units. For example, when a single straight track will serve the excavator, enough momentum may be built up by empty cars in descending the incline to carry them over a considerable length of level track to the excavator.

Both tail-rope and endless-rope haulage systems may be operated around curves, over undulating track, and on branches from the main haulage route. Neither has found as favorable reception in American practice as locomotives, but both have been highly developed in European coal-mine practice.

Structural Limitations

The simple engine plane can be used only on grades that impart enough velocity to the descending unit to overcome the resistance of car or train plus the frictional resistance of the dragging rope. As they have only a single track, their service is intermittent in that the loaded car or train must be hauled to the dump and returned before a second load can be attached. They can be used to serve several excavation points over various tracks branching from the foot of the incline, if the momentum is sufficient to carry the descending cars to the excavator. In such instances, the haulage rope is cut in several pieces, one for each excavation point. As an empty car or train is delivered to one excavation point, the rope is left attached to the car or train but is disconnected at the foot of the incline. A rope from another point is then attached to that on the incline, and loads are hauled from that point. Service is thus intermittent from each digging point, although it may be continuous from the pit as a whole. This intermittent service is not suitable for use with mechanical excavators but may be advantageous for hand loading when the time required to load car or train equals the haulage cycle from another loading point in the pit.

When the engine plane is used to supplement other haulage units its intermittent operation is of less consequence, since the supplemented unit supplies continuous service to the excavators.

The engine plane may haul the pit cars singly or in trains as delivered to it, or it may use only 1 or 2 cars of different design into which the pit cars may be dumped.

Engine planes are seldom used except to haul loads up grades too steep for locomotives. They may be used on any grade from, say, 6 percent to a little less than vertical. On grades of over 45°, however, they usually require especially designed cars or skips to prevent spillage.

While single-track, simple engine planes are sometimes employed, they are usually limited to small production. Ordinarily, the double-track plane is used with balanced hoisting.

The gravity plane requires grades too severe for locomotive haulage and is used with double-track balanced movement only to lower loaded cars or trains down grade. If a simple brake wheel is used, no power is available to hoist cars or supplies to the working face other than that furnished by the descending loads. If a hoist is used power can be applied independently of the descending loads when necessary. Whatever the device, it must be located at the top of the incline and cannot follow the excavator. Therefore, gravity planes are limited to use in supplementing some other type of haulage unit serving the excavator.

Gravity planes may lower pit cars singly or in trains, or they may use especially designed cars as does the engine plane. They can be used for any grade in which the pull of the descending load is enough to overcome the friction of both descending and ascending cars, the frictional resistance of both trailing ropes, and the grade resistance of the ascending empty cars.

Like the engine plane its service is intermittent, but since it is used only to supplement other units this may be of minor importance. Under proper conditions, however, the use of a gravity plane effectually reduces haulage costs, since no power input is required. It can be used on the same range of grades as the engine plane.

Tail-rope and endless-rope haulage systems are seldom used in gravel pits. While they may be used on level or undulating tracks and around curves, they are usually limited to grades of less than 45°.

Service Equipment Required

Any haulage system using a hoist for motive power requires a rope as the connecting link between hoist and car or train. Ropes for haulage systems are composed of wires wound into strands, the strands then being wound into the finished rope. Wire rope may be purchased in various combinations in which the number of wires in the strands and the number of strands in the rope both vary.

Standard "hoisting" rope is composed of 6 strands, each having 19 wires. This type of rope is built primarily for vertical hoisting in which the rope does not drag. However, they are commonly used in engine-plane duty.

Standard "haulage" rope is composed of 6 strands, but each strand has only 7 wires. This type of rope is built for engine-plane and inclined-shaft hoisting in which the rope is subject to abrasion from dragging.

Standard hoisting rope is more flexible and somewhat stronger than standard haulage rope of equal diameter. Both types are made with either hemp or wire-rope centers. The former are more flexible and are more commonly used.

Haulage rope requires sheaves and hoist drums of larger diameter than hoisting rope. Sheave and drum diameters recommended are as follows:

	<u>Minimum</u>	<u>Average</u>
For 6 x 7 haulage ropes multiply rope diameter by....	42	72
For 6 x 19 hoisting ropes multiply rope diameter by	30	45

Both hoisting and haulage ropes may be obtained in which the wires are made of various grades of material ranging from low to high tensile strength as follows: Cast steel, plow steel, and improved plow steel. Plow steel is more commonly used for haulage ropes.

Haulage ropes may also be purchased in which the method of winding wires and strands varies. In "regular-lay" rope the wires in the strands are wound in one direction and the strands in the opposite direction. Thus a rope in which the wires are wound to the left and the strands to the right is known as a "right-lay" rope and one in which the wires wind to the right and the strands to the left as a "left-lay" rope. Both are regular lay. Ropes are also made with both wires and strands winding in the same direction. Those are known as "lang"-lay ropes. Lang-lay rope is somewhat more flexible than regular lay, and since the wires are exposed at the surface of the rope in longer lengths it will stand greater wear or abrasion and is therefore more suitable for haulage ropes that drag on the ground or over rollers. It untwists more readily than regular lay, however, and is apt to kink when tension is removed. For this reason it is not as easy to handle in coupling cars. Table 51 presents the principal characteristics of standard hoisting ropes and table 52 those for standard haulage ropes.

For haulage purposes the breaking strength of the rope should be 5 to 7 times the stress at which it is used. The safety factor of 5 should be used for steep inclines and that of 7 for low grades where the rope drags heavily.

The single- or double-rope engine plane ordinarily serves some other type of haulage unit and seldom requires service equipment other than the ropes. The hoist is stationary and requires fuel service, and this will depend upon the type of power used. In installations in which the plane hoists pit cars, no auxiliary service equipment is required. Planes employing especially designed cars or skips, into which pit cars are dumped, require some sort of apparatus for receiving the load from the pit car and transferring it to the car on the incline. Both types may require hoppers into which the loads are dumped at the top of the incline.

TABLE 51.- Characteristics of standard hoisting rope:
6 strands, 19 wires, hemp center

Diameter of rope, inches	Approximate weight per foot, pounds	Recommended diameter of sheave, feet	Approximate breaking strength, tons of 2,000 pounds		
			Cast steel	Plow steel	Improved, plow steel
3/8.....	0.23	1.5	4.5	5.5	6.3
1/2.....	.40	2.0	7.7	9.4	10.8
5/8.....	.63	2.5	11.8	14.4	16.6
3/4.....	.90	3.0	16.8	20.6	23.7
7/8.....	1.23	3.5	22.8	28.0	32.2
1.....	1.60	4.0	29.5	36.5	42.0
1 1/8.....	2.03	4.5	37.0	46.0	53.0
1 1/4.....	2.50	5.0	46.0	56.5	65.0
1 3/8.....	3.03	5.5	55.0	68.0	78.5
1 1/2.....	3.60	6.0	65.0	80.5	92.5
1 5/8.....	4.23	6.5	76.0	94.0	108.0
1 3/4.....	4.90	7.0	88.0	108.0	124.0
1 7/8.....	5.63	8.0	100.0	123.0	142.0
2.....	6.40	8.0	114.0	140.0	161.0

TABLE 52.- Characteristics of standard haulage ropes:
6 strands, 7 wires, hemp center

Diameter of rope, inches	Approximate weight per foot, pounds	Recommended diameter of sheave, feet	Approximate breaking strength, tons of 2,000 pounds		
			Cast steel	Plow steel	Improved, plow steel
3/8.....	0.21	2.25	4.3	5.15	5.9
1/2.....	.38	3.0	7.5	9.0	10.3
5/8.....	.59	4.0	11.5	13.8	16.0
3/4.....	.84	4.5	16.5	19.8	22.8
7/8.....	1.15	5.5	22.4	26.8	30.8
1.....	1.50	6.0	29.0	34.8	40.0
1 1/8.....	1.90	7.0	36.4	43.6	50.0
1 1/4.....	2.34	7.5	44.5	53.0	61.0
1 3/8.....	2.84	8.5	53.5	63.5	73.5
1 1/2.....	3.38	9.0	62.5	75.0	86.5

Where special incline cars are used the lower end of the incline track is frequently sunk below the pit floor. Pit cars are then run on trestles or bridges over, and dumped directly into, the incline car. In more elaborate systems, the pit cars pass over a wood, steel, or concrete hopper or bin. Delivery from bin or hopper may be directly through chutes controlled by gates or over feeder conveyors.

Capacity

There is no standardization of hoists and rope-haulage units. Hoists for such purposes are obtainable in almost any size and type using any kind of power and for any haulage duty. The hoists themselves are standardized by different manufacturers, and more or less uniformity exists between those standards. By this it is meant that manufacturers standardize their equipment by horsepower rating or the pounds of pull or speed the hoist can impart to the rope, although the design of the machine itself will vary with each factory, and in fact a single factory may produce various designs of equal power rating, rope pull, or speed.

Horsepower rating does not apply directly to the hoist itself but refers to the power unit used to drive the hoist. Custom, however, has made it common practice to refer to a hoist as of a certain horsepower. Strictly interpreted, this means that the particular hoist was designed to utilize the power developed by a motor or engine of the designated horsepower.

Rating by rope pull or speed refers directly to the hoist itself in translating the power of its prime mover into motion. Both the rope pull and speed of the hoist depend, however, upon the power and speed of the prime mover and the connecting arrangement between motor and hoist. Hence, all three ratings should be known to get a complete picture of the duty characteristics of any hoist.

Ordinarily the speed of the driving engine or motor, as expressed in revolutions per minute, is greater than that required in the hoist drum, and some method of speed reduction must be introduced. This may be done by different pulley diameters with belt or rope drive or by different tooth ratios with chain or gear drive. Sometimes reduction is only partly accomplished by this means, and further reduction occurs in the mechanism of the hoist itself.

In some instances, the revolutions per minute of the motive power and hoist drum are the same, and they are mounted on the same shaft or connected through rigid or flexible couplings.

The selection of a hoist for any type of rope haulage system requires information as to the rope pull and speed of which it is capable and the horsepower and speed required in the motor or engine to produce that rope pull and speed.

The rope pull required of a hoist for engine-plane duty will depend upon the grade of the plane, the gross or net weight and resistance of car or train (single or balanced hoisting), the frictional resistance of the dragging rope or ropes, and the speed of acceleration required.

The rope speed required for a similar duty will depend upon the length of haul or inclined length of plane and the relation between the payload capacity of the car or train and the time cycle permissible.

The horsepower necessary in the driving unit to produce the required rope pull and speed will depend upon the rope pull at the hoist drum, the efficiency of power transmission between drive unit and hoist, and to some extent on the type of power used.

Obviously the multiplicity of requirements due to variation in these factors prevents any tabulation of the capacities of hoist and rope-haulage units. While it is true that engine planes are seldom used, nevertheless where they are necessary it is important that they have adequate capacity when installed; therefore, a method of calculation is presented as follows:

Figure 18 represents an inclined plane of any grade on which rests a body indicated by the circle. The weight of this body due to the force of gravity acts vertically downward in the direction of the arrow pointing to W. However, since it rests on the plane, it cannot move in that direction. This force can be resolved into its two components, one acting parallel to the plane in the direction of A and one acting perpendicular to the plane in the

direction B. The former causes motion down the plane and the latter causes friction with the plane. To prevent motion in the body a force must be supplied acting in the direction T and equal to that acting toward A.

The force acting toward A is computed as $W \sin a$ in which W equals the weight of the body and a equals the angle the plane makes with the horizontal. The force acting toward B is computed as $W \cos a$. The angular range of an inclined plane is from level or 0° to vertical or 90° . The sine of an angle of 1° equals the cosine of an angle of 89° , and their values are complementary throughout this range. Table 53 has been compiled from this relationship, in which the weight W is assumed as 100 units and the values of the forces acting toward A and B are computed for each degree of the angular range. The values thus represent a percentage of weight W .

For example, a weight of 100 pounds on a plane of 18° will exert (from the table) a force of 30.9 pounds downward parallel to the plane (A) and 95.11 pounds normal to the plane (B). In other words, on an 18° plane the force tending to move a body down the plane is 30.9 percent of its weight, and the force acting to produce friction is 95.11 percent of that weight.

In order to hoist the body up the incline a force must be applied in direction T equal to that acting in the direction A plus the friction caused by the force acting toward B.

The weight of the body on the plane is represented by the sum of the weight of the empty car or train, the weight of the load carried by car or train, and the weight of the haulage rope. The car and load weights will remain constant for the entire length of the plane, but the weight of the haulage rope will vary as the car is moved. It will be a maximum when the loaded car is at the bottom of the incline and a minimum when the car is at the top. Since the capacity of the hoist must be computed when the load is the greatest, the weight of the rope should be taken at the maximum.

The friction caused by the force acting toward B affects both the resistance of the car and its load and the haulage rope. The car friction can be computed the same as the friction of a car used with a locomotive as the motive power, except that the weight of the car will vary with the force acting to produce friction, which varies as the cosine of the plane angle.

The friction of the haulage rope will depend upon the length of rope in contact with the plane, the nature of the plane surface, and whether idler rollers are used to keep the rope above the plane surface. The length of rope in contact with the plane will depend upon the sag of the rope, which in turn is governed by the span, tension, and weight of the rope per foot of length, and the height above the plane surface of the point of attachment to the car and upper sheave. Obviously local conditions will govern the amount of rope friction, and computations must be limited to each particular installation.

The sag in horizontal haulage ropes may be computed from the following formula:

$$D = \frac{S^2 W}{8 T},$$

in which

D = deflection or sag, in feet;

S = length of span, in feet;

W = weight of the rope per foot, in pounds; and

T = tension on the rope, in pounds.

TABLE 53.- Values of A and B in percent of gross weight

A, degrees	Percent of W	B, degrees	A, degrees	Percent of W	B, degrees	A, degrees	Percent of W	B, degrees
Level	0.00	90						
1	1.75	89	31	51.50	59	61	87.46	29
2	3.49	88	32	52.99	58	62	88.30	28
3	5.23	87	33	54.46	57	63	89.10	27
4	6.98	86	34	55.92	56	64	89.88	26
5	8.72	85	35	57.36	55	65	90.63	25
6	10.45	84	36	58.78	54	66	91.36	24
7	12.19	83	37	60.18	53	67	92.05	23
8	13.92	82	38	61.57	52	68	92.72	22
9	15.64	81	39	62.93	51	69	93.36	21
10	17.37	80	40	64.28	50	70	93.97	20
11	19.08	79	41	65.61	49	71	94.55	19
12	20.79	78	42	66.91	48	72	95.11	18
13	22.50	77	43	68.20	47	73	95.63	17
14	24.19	76	44	69.47	46	74	96.13	16
15	25.88	75	45	70.71	45	75	96.59	15
16	27.56	74	46	71.93	44	76	97.03	14
17	29.24	73	47	73.13	43	77	97.44	13
18	30.90	72	48	74.31	42	78	97.82	12
19	32.56	71	49	75.47	41	79	98.16	11
20	34.20	70	50	76.60	40	80	98.48	10
21	35.84	69	51	77.71	39	81	98.77	9
22	37.46	68	52	78.80	38	82	99.03	8
23	39.07	67	53	79.86	37	83	99.26	7
24	40.67	66	54	80.90	36	84	99.45	6
25	42.26	65	55	81.91	35	85	99.62	5
26	43.84	64	56	82.90	34	86	99.76	4
27	45.40	63	57	83.87	33	87	99.86	3
28	46.95	62	58	84.80	32	88	99.94	2
29	48.48	61	59	85.72	31	89	99.98	1
30	50.00	60	60	86.60	30	90	100.00	Level

Table 54 has been computed from this formula to show the sag for various spans and various amounts of tension for a rope weighing 1 pound per foot. The sag for other ropes may be computed by multiplying the figures given in the table by the weight per foot of the rope.

Tables 51 and 52 give the approximate weights per foot and breaking strengths of ropes of various diameters.

The sag in haulage ropes used on planes in which one end of the span is higher than the other will vary somewhat from that calculated by the formula, but the variation will be insufficient to affect seriously the computation for sand and gravel haulage.

TABLE 54.- Sag in feet for wire rope weighing 1 pound per foot

Span, feet	Tension, pounds											
	100	200	300	400	500	1,000	5,000	10,000	20,000	30,000	40,000	50,000
100	12.5	6.25	4.17	3.13	2.50	1.25	0.25	0.13	0.06	0.04	0.03	0.03
150	28.1	14.05	9.37	7.00	5.63	2.81	.56	.28	.14	.09	.07	.06
200	50.0	25.00	16.67	12.50	10.00	5.00	1.00	.50	.25	.17	.13	.10
250	78.1	39.05	26.03	19.50	15.62	7.81	1.56	.78	.39	.26	.20	.16
300	112.5	56.25	37.50	28.25	22.50	11.25	2.25	1.13	.56	.37	.28	.23
350		76.60	51.10	38.80	30.60	15.30	3.06	1.53	.77	.51	.39	.30
400		100.00	66.67	50.00	40.00	20.00	4.00	2.00	1.00	.67	.50	.40
450		126.50	84.40	63.25	50.60	25.30	5.06	2.53	1.27	.84	.63	.50
500		156.25	104.20	78.13	62.50	31.25	6.25	3.13	1.56	1.04	.78	.62
550		189.00	126.00	94.50	75.60	37.80	7.56	3.78	1.89	1.26	.95	.75
600				112.50	90.00	45.00	9.00	4.50	2.25	1.50	1.13	.90
650				132.00	105.60	52.80	10.56	5.28	2.64	1.76	1.32	1.06
700				153.00	122.50	61.25	12.25	6.13	3.06	2.04	1.53	1.23
750				176.00	140.60	70.30	14.06	7.03	3.51	2.34	1.76	1.41
800				200.00	160.00	80.00	16.00	8.00	4.00	2.70	2.00	1.60
850						90.30	18.06	9.03	4.51	3.01	2.25	1.80
900						101.00	20.20	10.10	5.05	3.37	2.52	2.02
950						113.00	22.60	11.30	5.56	3.80	2.78	2.26
1,000						125.00	25.00	12.50	6.25	4.17	3.13	2.50
1,100						151.00	30.20	15.10	7.55	5.03	3.77	3.02
1,200						180.00	36.00	18.00	9.00	6.00	4.50	3.60
1,300						211.00	42.20	21.10	10.55	7.03	5.27	4.22
1,400						245.00	49.00	24.50	12.25	8.17	6.13	4.90
1,500						281.00	56.20	28.10	14.05	9.37	7.03	5.62
1,600						320.00	64.00	32.00	16.00	10.67	8.00	6.40
1,700							72.25	36.13	18.06	12.04	9.03	7.23
1,800							81.00	40.50	20.25	13.50	10.13	8.10
1,900							90.30	45.15	22.57	15.05	11.28	9.03
2,000							100.00	50.00	25.00	16.67	12.50	10.00
3,000							225.00	112.50	56.25	37.50	28.13	22.50

The approximate length of rope which drags on the plane can be calculated from the sag as found for any particular installation. The height of the car coupling varies with the car design, ranging from 12 to 34 1/2 inches and averaging roughly 24 inches. The height of the upper sheave above the ground varies with each installation. This sheave may be installed below the rails so that the car can run over it, or it may be in a building or on a structure above the track allowing the car to run under it. Customarily the sheave is placed below the rails so that the rope passes level with the top of the rail. In such instances, if the tension is sufficient the rope when starting the loads will assume a position in a straight line between sheave and car coupling and will not drag for any portion of its length. This, of course, is an extreme condition not met in practice. If the average height of the car coupling is 24 inches and the sheave is at the ground surface, the maximum sag permissible without dragging will be 12 inches. If the weight per foot, span, and tension are known for any particular installation, the sag can be calculated from table 54, and from the basic formula,

$$D = \frac{S^2 W}{8 T},$$

The difference between the span for a 1-foot sag and the actual span will give the approximate dragging length at the start of the trip. The dragging length will be reduced as the car moves up the plane, and consequently the rope friction will be reduced.

The car or train resistance will depend upon the condition of the cars and track and will range from a minimum of 10 pounds per ton of gross car weight for specially built plane cars in first-class condition to 50 pounds or more for poorly maintained pit cars.

The frictional resistance of the rope will depend upon the type and spacing of idler rollers, if used, or the condition of the roadbed or ties on which the rope drags. For the most efficient idlers the resistance may be as low as 100 pounds per ton of rope weight, and for dragging over rough, poorly ballasted ties it may be as great as 1,000 pounds per ton of rope weight.

Table 55 gives the friction and gravity components of unit car and rope weights for various angles of plane inclination.

TABLE 55.—Gravity and friction components of loads on inclines

Plane inclination		Rope tension due to gravity from 2,000 pounds of total load (car+gravel + rope), pounds	Rope tension due to resistance of 2,000 pounds of gross car weight (car+gravel), pounds		Rope tension due to friction of 2,000 pounds of rope weight, pounds	
			10 pounds per ton	50 pounds per ton	100 pounds per ton	1,000 pounds per ton
Degrees	Percent					
1	1.75	35	10	50	100	1,000
2	3.5	70	10	50	100	999
3	5.2	105	10	50	100	999
4	7.0	140	10	50	100	998
5	8.7	174	10	50	100	996
6	10.5	209	10	50	100	995
7	12.3	244	10	50	99	993
8	14.0	278	10	50	99	990
9	15.8	312	10	49	99	988
10	17.6	347	10	49	99	985
11	19.4	382	10	49	98	982
12	21.3	416	10	49	98	978
13	23.1	450	10	49	97	974
14	25.0	484	10	49	97	970
15	26.8	518	10	48	97	966
16	28.7	551	10	48	96	961
17	30.6	584	10	48	96	956
18	32.5	618	10	48	95	951
19	34.4	651	9	47	95	946
20	36.4	684	9	47	94	940
21	38.4	716	9	47	93	934
22	40.4	749	9	46	93	927
23	42.4	781	9	46	92	921

TABLE 55.- Gravity and friction components of loads on inclines - Continued

Plane inclination		Rope tension due to gravity from 2,000 pounds of total load (car+gravel + rope), pounds	Rope tension due to resistance of 2,000 pounds of gross car weight (car+gravel), pounds		Rope tension due to friction of 2,000 pounds of rope weight, pounds	
Degrees	Percent		10 pounds per ton	50 pounds per ton	100 pounds per ton	1,000 pounds per ton
24	44.5	814	9	46	91	914
25	46.6	845	9	45	91	906
26	48.8	877	9	45	90	899
27	51.0	908	9	45	89	891
28	53.2	939	9	44	88	883
29	55.4	970	9	44	88	875
30	57.7	1,000	9	43	87	866
31	60.0	1,030	9	43	86	857
32	62.5	1,060	8	42	85	848
33	64.9	1,089	8	42	84	839
34	67.5	1,118	8	41	83	829
35	70.0	1,147	8	41	82	819
36	72.7	1,176	8	40	81	809
37	75.4	1,204	8	40	80	799
38	78.1	1,231	8	39	79	788
39	81.0	1,258	8	39	78	777
40	83.9	1,286	8	38	77	766
41	86.9	1,312	8	38	76	755
42	90.0	1,338	7	37	74	743
43	93.2	1,364	7	37	73	731
44	96.6	1,390	7	36	72	719
45	100	1,414	7	35	71	707
46	103	1,439	7	35	70	695
47	107	1,463	7	34	68	682
48	111	1,486	7	33	67	669
49	115	1,510	7	33	66	656
50	119	1,532	6	32	64	643
51	123	1,554	6	31	63	629
52	128	1,576	6	31	62	616
53	133	1,597	6	30	60	602
54	138	1,618	6	29	59	588
55	143	1,638	6	29	57	574
56	148	1,658	6	28	56	559
57	154	1,678	5	27	55	545
58	160	1,696	5	27	53	530
59	166	1,714	5	26	52	515
60	173	1,732	5	25	50	500

TABLE 55.- Gravity and friction components of loads on inclines - Continued

Plane inclination		Rope tension due to gravity from 2,000 pounds of total load (car+gravel + rope), pounds	Rope tension due to resistance of 2,000 pounds of gross car weight (car+gravel), pounds		Rope tension due to friction of 2,000 pounds of rope weight, pounds	
Degrees	Percent		10 pounds per ton	50 pounds per ton	100 pounds per ton	1,000 pounds per ton
61	180	1,750	5	24	49	485
62	188	1,766	5	24	47	469
63	196	1,782	5	23	45	454
64	205	1,798	4	22	44	438
65	214	1,813	4	21	42	423
66	225	1,827	4	20	41	407
67	236	1,841	4	20	39	391
68	247	1,854	4	19	38	375
69	260	1,867	4	18	36	358
70	275	1,880	3	17	34	342
71	290	1,891	3	16	33	326
72	308	1,902	3	15	31	309
73	327	1,913	3	15	29	292
74	349	1,922	3	14	28	276
75	373	1,932	3	13	26	259
76	401	1,941	2	12	24	242
77	433	1,949	2	11	23	225
78	470	1,956	2	10	21	208
79	514	1,963	2	10	19	191
80	567	1,970	2	9	17	174
81	631	1,976	2	8	16	156
82	711	1,981	1	7	14	139
83	814	1,985	1	6	12	122
84	951	1,989	1	5	11	105
85	1,143	1,992	1	4	9	87
86	1,430	1,995	1	3	7	70
87	1,908	1,997	1	3	5	52
88	2,864	1,998	0	2	4	35
89	5,729	1,999	0	1	2	17
90	Infinity	2,000	0	0	0	0

Besides the tension due to gravity and the car and rope frictional resistances, the haulage rope is subject to additional tension caused by acceleration.

Acceleration is usually expressed as an increase of speed in feet per second for each second acceleration lasts. It may be computed from the formula:

$$a = v/t$$

in which

v = velocity in feet per second,

a = acceleration in feet per second per second, and

t = time in seconds.

The tension due to acceleration can be computed from the formula:

$$F = Wa/g$$

in which

W = gross weight of car, load, and rope, pounds;

a = acceleration in feet per second per second;

g = acceleration due to gravity = 32.16; and

F = acceleration tension, pounds.

Combining the two formulas,

$$F = Wv/gt$$

Table 56 has been computed to present the accelerative force in pounds per ton of gross load for various accelerative periods and final velocities.

TABLE 56.- Accelerative force required for 2,000 pounds gross load, pounds

Acceleration period, seconds	Velocity, feet per minute												
	100	200	300	400	500	600	700	800	900	1,000	1,250	1,500	2,000
5	20.7	41.5	62.2	83.0	104.0	124.0	145.0	166.0	187.0	207.0	259.0	311.0	415.0
6	17.3	34.5	51.8	69.1	86.4	104.0	121.0	138.0	155.0	173.0	216.0	259.0	345.0
7	14.8	29.6	44.4	59.2	74.0	88.9	104.0	118.0	133.0	148.0	185.0	222.0	296.0
8	13.0	25.9	38.9	51.8	64.8	77.8	90.7	104.0	117.0	130.0	162.0	194.0	259.0
9	11.5	23.0	34.6	46.1	57.6	69.1	80.7	92.2	104.0	115.0	144.0	173.0	230.0
10	10.4	20.7	31.1	41.5	51.9	62.2	72.6	83.0	93.4	104.0	130.0	156.0	207.0
15	6.9	13.8	20.7	27.6	34.5	41.5	48.4	55.3	62.2	69.1	86.3	104.0	138.0
20	5.2	10.4	15.5	20.7	25.9	31.1	36.3	41.5	46.6	51.8	64.8	77.7	104.0
25	4.2	8.3	12.5	16.6	20.7	24.9	29.1	33.2	37.4	41.5	51.9	62.2	83.0
30	3.5	6.9	10.4	13.8	17.3	20.7	24.2	27.7	31.1	34.6	43.3	51.9	69.2
35	3.0	5.9	8.9	11.8	14.8	17.8	20.7	23.7	26.6	29.6	37.0	44.4	59.2
40	2.6	5.2	7.8	10.4	13.0	15.5	18.1	20.7	23.3	25.9	32.4	38.9	51.8
45	2.3	4.6	6.9	9.2	11.5	13.8	16.1	18.4	20.7	23.0	28.8	34.5	46.0
50	2.1	4.1	6.2	8.3	10.4	12.4	14.5	16.6	18.6	20.7	25.9	31.1	41.4
55	1.9	3.8	5.6	7.5	9.4	11.3	13.2	15.0	16.9	18.8	23.5	28.2	37.6
60	1.7	3.5	5.2	6.9	8.7	10.4	12.1	13.8	15.6	17.3	21.6	26.0	34.6

The distance the car travels during the accelerative period may be computed from the following formula:

$$S = vt/2$$

in which

S = distance traveled in feet,
 v = final velocity in feet per second, and
 t = acceleration time period in seconds.

Table 57 has been compiled to present the distance traversed by a car for various accelerative periods and final velocities.

The horsepower required in engine-plane duty is a maximum when the loaded car is being started and accelerated at the foot of the incline. Horsepower may be computed from the following formula:

$$HP = TV/33,000$$

in which

T = total tension in pounds, including
acceleration stress; and
 V = final velocity in feet per minute.

From these formulas and tables it is believed that the capacity of any plane can be computed or the equipment designed to fit a desired capacity. Two examples have been worked out for illustration.

TABLE 57.- Distance covered in feet for various accelerative periods and final velocities
 $(s = vt/2)$; initial velocity = 0

Acceler- ation period seconds	Final velocity, feet per minute												
	100	200	300	400	500	600	700	800	900	1,000	1,250	1,500	2,000
5	4.17	8.3	12.5	16.7	20.8	25.0	29.2	33.3	37.5	41.7	52.1	62.5	83.3
6	5.0	10.0	15.0	20.0	25.0	30.0	35.0	40.0	45.0	50.0	65.0	75.0	100
7	5.83	11.7	17.5	23.3	29.2	35.0	40.8	46.6	52.5	58.3	72.9	87.5	117
8	6.67	13.3	20.0	26.7	33.3	40.0	46.7	53.3	60.0	66.7	83.3	100	133
9	7.50	15.0	22.5	30.0	37.5	45.0	52.5	60.0	67.5	75.0	93.8	113	150
10	8.33	16.7	25.0	33.3	41.7	50.0	58.3	66.7	75.0	83.3	104	125	167
15	12.50	25.0	37.5	50.0	62.5	75.0	87.5	100	113	125	156	188	250
20	16.67	33.3	50.0	66.7	83.3	100	117	133	150	167	208	250	333
25	20.83	41.7	62.5	83.3	104	125	146	167	188	208	260	313	417
30	25.00	50.0	75.0	100	125	150	175	200	225	250	313	375	500
35	29.17	58.3	87.5	117	146	175	204	233	263	292	365	438	583
40	33.33	66.7	100	133	167	200	233	267	300	333	417	500	667
45	37.50	75.0	113	150	188	225	263	300	338	375	469	563	750
50	41.67	83.3	121	167	208	250	292	333	375	417	521	625	833
55	45.83	91.7	138	183	229	275	321	367	413	458	573	688	917
60	50.00	100	150	200	250	300	350	400	450	500	625	750	1,000

Example 10.— Assume that an operator wishes to install a simple engine plane to hoist 1,000 tons per day of 8 hours from his pit through a vertical distance of 100 feet and that the plane length must approximate 300 feet in horizontal distance. Pit cars will be hoisted singly and weigh 4 tons each, with a capacity of 6 cubic yards or 9 tons.

$$\frac{1,000}{8} = 125 \text{ tons hourly capacity.}$$

$$\frac{125}{9} = 14 \text{ carloads hourly or a round trip on the plane of}$$

$$\frac{60}{14} = 4.3 \text{ or, say, 4 minutes.}$$

If 30 seconds is allowed to dump the car and a like interval to change the rope from one car to another at the foot of the incline, the traveling time per trip must be 3 minutes, or 1.5 minutes each way, assuming equal speed up and down.

The plane has an inclination of approximately 33 1/3 percent or, say, 18°. From trigonometry the inclined length equals the vertical height divided by the sine of the angle. Table 53 shows trigonometric sines and cosines of angles represented as percent. The sine of 18° from the table is 0.3090. Then

$$\frac{100}{0.3090} = 324 \text{ feet, or the inclined length of the plane.}$$

The horizontal length, again from trigonometry, equals the inclined length multiplied by the cosine of the angle. From table 53 the cosine of 18° is 0.9511. Then

$$324 \times 0.9511 = 308 \text{ feet, or the horizontal length.}$$

Since this is only slightly greater than the 300 feet assumed as limiting the horizontal length, the plane will be considered as inclined at an angle of 18° and as 324 feet long.

The car must then travel 324 feet in 1.5 minutes or 90 seconds, or at an average speed of 216 feet per minute.

Assuming that the acceleration period is 10 seconds and that the time required for deceleration is the same, the car will be traveling at constant speed for 90 - 10 - 10 = 70 seconds.

From table 57 a car starting from rest and attaining a speed of 250 feet per minute in 10 seconds will have traveled $\frac{25 + 16.7}{2} = 20.8$ feet. Then 324 - 20.8 - 20.8 = 282.4 feet is

to be covered at constant speed in 70 seconds, or 242 feet per minute. Therefore, the hoist should be capable of a drum speed of 250 feet per minute.

The gross weight of the loaded car is

	<u>Tons</u>
Car.....	4
Gravel	<u>9</u>
Total	13

If the weight of the rope is disregarded for the moment, reference to table 55 shows that the rope tension due to gravity of 1 ton on an 18° plane is 618 pounds and the car re-

sistance (at 50 pounds per ton) is 48 pounds, or a total tension due to car and load of 666 pounds. From table 56 the accelerative force required to bring 1 ton of load to a speed of 250 feet per minute in 10 seconds will be $\frac{20.7 + 31.1}{2} = 25.9$ pounds. Hence the total tension

due to the load is $(618 + 48 + 25.9) \times 13$ tons = 8,995 pounds or 4.5 tons.

As this is a fairly flat plane a factor of safety of 7 is required. The rope then should have a minimum breaking strength of $4.5 \times 7 = 31.5$ tons. From table 52, a 1-inch, plow-steel haulage rope has a breaking strength of 34.8 tons or more than the minimum required. Therefore a 1-inch rope is selected.

The tension on the hoist can then be computed as follows:

	<u>Pounds</u>
Car weight.....	8,000
Gravel weight.....	18,000
324 feet of 1-inch rope	<u>488</u>
Gross load.....	26,488

Reference to table 54 shows that a rope weighing 1 pound per foot under 10,000 pounds tension on a 350-foot span will sag 1.53 feet. Therefore a 1-inch rope weighing 1.5 pounds per foot may be calculated to sag 1.5×1.53 or 2.25 feet. The span required for a 1-foot sag is

$$S = \sqrt{\frac{8 \cdot DT}{W}} = \sqrt{\frac{8 \times 1 \times 10,000}{488}} = 164 \text{ feet.}$$

$$324 - 164 = 160 \text{ feet of rope dragging.}$$

$$160 \times 1.5 = 240 \text{ pounds, weight of dragging rope.}$$

From tables 55 and 56, the various tensions are as follows:

	<u>Pounds</u>
Gravity, $\frac{26,488}{2,000} \times 618$	= 8,180
Car resistance, $\frac{26,000}{2,000} \times 48$	= 624
Rope friction, $\frac{240}{2,000} \times 951$	= 114
Acceleration, $\frac{26,488}{2,000} \times 25.9$	= <u>343</u>
Maximum tension.....	9,261

At the minimum resistance offered by cars in best condition and with adequate rope idlers the tension would be

	<u>Pounds</u>
Gravity.....	= 8,180
Car resistance, $\frac{26,000}{2,000} \times 10$	= 130
Rope resistance, $\frac{240}{2,000} \times 95$	= 11
Acceleration.....	= <u>343</u>
Total.....	8,664

This maximum tension is exerted for only the acceleration period or 10 seconds. Hence the indicated horsepower required is

$$\text{I.hp.} = \frac{\text{TV}}{33,000} = \frac{9,261 \times 250}{33,000} = 70.2, \text{ and}$$

$$\text{I.hp.} = \frac{8,664 \times 250}{33,000} = 65.6.$$

At 75-percent efficiency the power required would be 88.5 to 94 hp.

From table 52 the hoist drum should be 6 feet in diameter for a 1-inch haulage rope, but for hoisting rope it might be 4 feet in diameter.

With haulage rope the drum speed would be 13.26 r.p.m., and with hoisting rope it would be 19.9 r.p.m. to obtain the required rope speed of 250 feet per minute.

The time cycle with such equipment will be as follows:

Hoisting:	Feet	Seconds
Acceleration....	20.8.....	10
Constant speed	282.4 at 250 f. p.m.	68
Deceleration....	20.8.....	10
Total.....	324.0.....	88
Dumping.....		30
Lowering (controlled by brake but assumed same as hoisting)		88
Changing rope.....		30
Total cycle (3 minutes 56 seconds).....		236

Then $9 \times \frac{3,600}{236} \times 8 = 1,100$ tons per day, or 10 percent more than required in the problem.

This margin is small but can easily be enlarged by releasing the brake and speeding up the downhill return of the empty car.

Example 11.— The same conditions are assumed as in example 10, except that two cars and ropes are used and hoisted in balance.

Starting with a loaded car at the bottom of the incline and an empty car at the top, the loads on the two ropes are as follows:

	Up trip, pounds	Down trip, pounds
Car weight.....	8,000	8,000
Gravel weight.....	18,000	
Rope weight.....	488	
Total.....	26,488	8,000

The tensions due to gravity and frictional resistances are

	<u>Up trip, pounds</u>	<u>Down trip, pounds</u>
Gravity, $\frac{26,488}{2,000} \times 618$ =	8,180	
$\frac{8,000}{2,000} \times 618$ =		2,472
Car resistance, $\frac{26,000}{2,000} \times 48$ =	624	
$\frac{8,000}{2,000} \times 48$ =		-192
Rope friction, $\frac{240}{2,000} \times 951$ =	114	0
Acceleration, $\frac{26,488}{2,000} \times 25.9$ =	343	
$\frac{8,000}{2,000} \times 25.9$ =		103.6
Total.....	9,261	2,383.6

The unbalanced pull on the hoist drum is then

$$9,261 - 2,383.6 = 6,877.4 \text{ pounds.}$$

The maximum indicated horsepower is

$$\text{I.hp.} = \frac{\text{TV}}{33,000} = \frac{6,877.4 \times 250}{33,000} = 52 \text{ hp.}$$

With best cars and idlers the unbalanced pull would be

$$8,664 - 2,535.6 = 6,128.4 \text{ pounds, and the indicated horsepower}$$

$$\frac{6,128.4 \times 250}{33,000} = 46.5.$$

At 75-percent efficiency the power required would range from 62 to 70 hp.

Balanced hoisting thus saves about 25 hp. or nearly 30 percent of that required for the simple engine plane.

The loaded car would be dumped while the rope was being changed at the bottom of the plane. Hence the time cycle would be the same as that for a single-track plane, but there would be 2 cars dumped instead of 1 car. The capacity would thus be doubled.

Remote-Control Haulage

Remote-control haulage or centrally controlled haulage systems have not, to the author's knowledge, been used in sand and gravel pits. They have been applied successfully to rock-quarry haulage, and since there is no fundamental reason why they should not be applied to gravel pits a brief discussion is included.

This type of haulage system consists of the operation by remote control of a number of self-propelled, burden-bearing cars by one man from a centrally located observation tower. Where local topography or construction prevents a view of the entire haulage route from one control point, 2 towers and 2 men are used.

The system is powered by electricity, the majority of installations using direct current supplied through a third or power rail. Recently alternating current has been utilized

successfully with two power rails. With either current the system is so designed that the entire haulage route is divided into several sections, any one or all of which may be energized at one time. Thus sections on which no cars are traveling may be cut out of the circuit automatically and may carry no current.

The cars used in these systems are built from designs made for each installation. Hence there is no standardization of design, except perhaps as to minor details. Each car consists of a cargo-carrying hopper up to 50 tons in capacity, which may be designed to dump to either side or end as desired, mounted on a heavy frame carried on a narrow- or standard-gage wheel base, usually of two 4-wheel trucks. The motive power is supplied by electric motors mounted on the trucks and geared to the axles. With direct current 2 to 4 motors are used, half the number driving the car in one direction and half in the other. Thus half the motors are idle with this type of current. With alternating current 2 motors are used, and the direction of travel is changed by reversing the direction of rotation of the motor through phase control. Cars using both types of current are fitted with brakes which automatically set when the current is cut off.

Haulage tracks with these systems can be either narrow or standard gage, and curves and grades suitable for locomotive haul can be negotiated.

Both systems are subject to the disadvantage of added difficulty and cost in shifting tracks when such shifts are necessary frequently to follow the moves of the excavator. This disadvantage is largely removed where it is possible to design the haulage route as a circle or ellipse.

Centrally controlled haulage systems are comparatively high in first cost and are not suited to small tonnage requirements. Where production is large enough the operating costs are reported to be less than those with locomotive haul.

These systems, like trolley locomotive systems, require well-maintained, electrically bonded tracks.

As they are controlled from 1 or 2 central towers, the operating labor is reduced to a minimum. Where the control tower is at such a distance that accurate spotting of the cars at the excavator becomes difficult, subcontrol stations can be set up to be operated by the shovel runner.

Operators of these systems claim that less rolling equipment is necessary with them than with locomotive haul and that because of more regularity in the time cycle less delay occurs at the shovel due to lack of haulage equipment.

The operation of centrally controlled systems is limited to large dry operations having a source of relatively cheap electric power.

For further details on this type of haulage the reader is referred to Bureau of Mines Information Circulars 6498 and 6513, describing the operations of the Trinity Portland Cement Co. at their Dallas and Fort Worth quarries, and to an article on pages 90-93 of the October 10, 1931 issue of Rock Products, entitled "New Quarry Haulage System Uses Alternating Current."

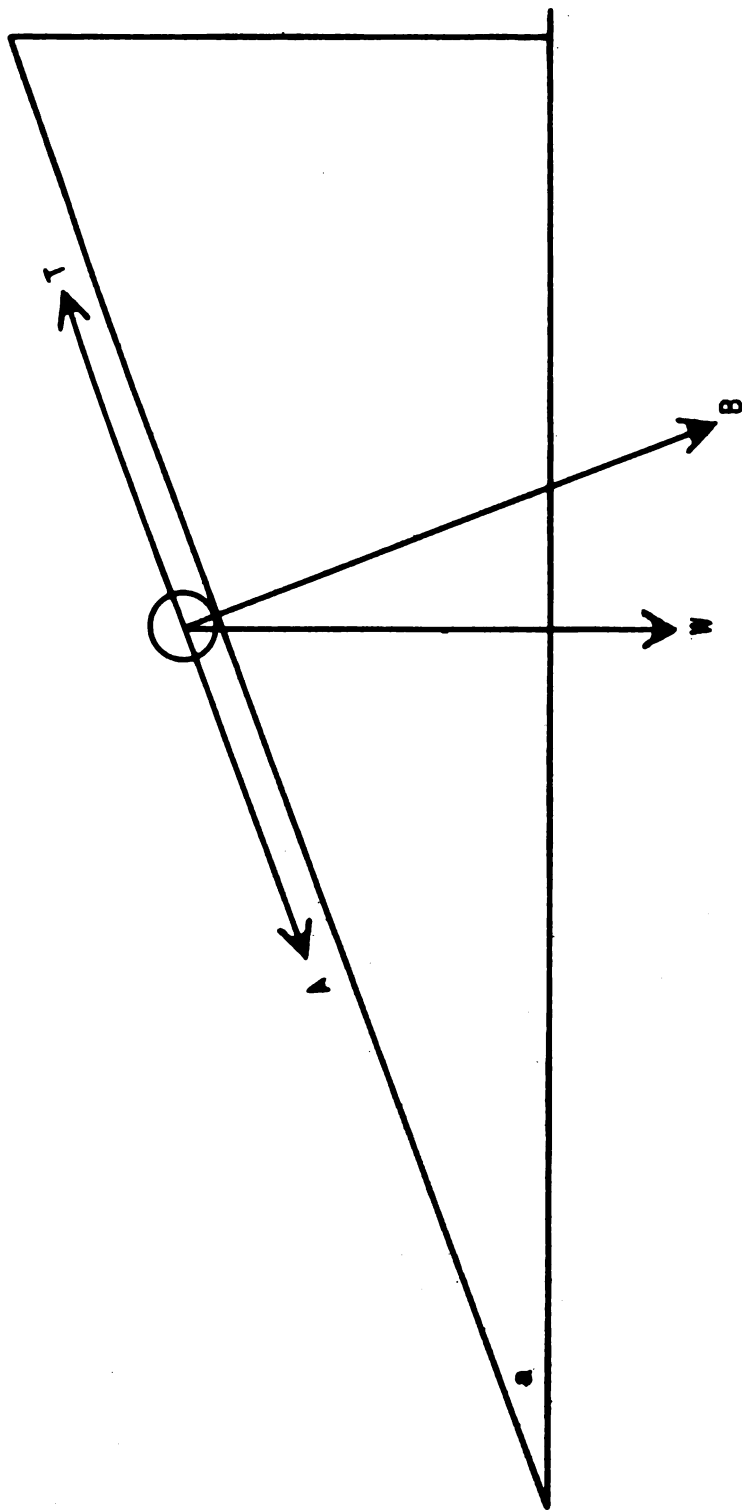


Figure 18.— Inclined plane.